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CONTRACT TRIAL OF THE UNITED STATES SEAGOING BATTLESHIP *ALABAMA*.

BY LIEUTENANT H. HALL, U. S. NAVY.

The *Alabama* is a twin-screw, armored, seagoing battleship, and was built by the Wm. Cramp and Sons' Ship and Engine Building Company, of Philadelphia, Pa., the general plans and specifications having been furnished by the Navy Department. The contract was signed September 24, 1896, the price being $2,650,000, and the time allowed for completion, three years. The price stated above does not include the armor, armor bolts, ordnance, ordnance outfit and a few articles supplied by the Government. The speed guarantee was sixteen knots per hour, to be maintained successfully for four consecutive hours, during which period the air pressure in the firerooms was not to exceed one inch of water, the vessel to be weighted to a mean draught of 23 feet 6 inches. No premium for excess of speed over that stipulated in the contract was provided for, but it was stipulated that a penalty at the rate of $25,000 per quarter knot between

sixteen and fifteen and a half knots be inflicted in case the speed fell below sixteen knots, and at the rate of $50,000 per quarter knot between fifteen and a half and fifteen knots, and in case the speed fell below fifteen knots it was optional with the Secretary of the Navy to reject the vessel or accept her at a reduced price. The weight of machinery was limited to 1,130 tons, including main engines, boilers and appurtenances, distilling apparatus, stores, spare parts, heating apparatus, tools in workshop, and water in boilers, condensers, pumps, pipes and stern tubes, but did not include turret machinery, capstan, windlass, steering gear and winches. The penalty for overweight was $500 per ton over 1,130 tons, and if this exceeded 5 per cent., then $10,000 further was to be deducted from the contract price of the vessel.

HULL.

The hull is constructed of mild steel of domestic manufacture, with frames spaced 4 feet apart, except where intermediate frames are placed, from the stem to frame No. 76; from this point aft they are spaced 3½ feet. The thickness of the outer keel plate is ¾ inch, of the inner, ⅝ inch, and of the vertical keel, ½ inch. The outer bottom plating is ⅝ inch thick, and the inner bottom plating ⅜ inch.

The ship is divided into two hundred and eighty-one watertight compartments. The double bottom extends from frame 22 to frame 75, a distance of 212 feet, and runs up to the armor shelf. It is 3 feet in depth amidships and is divided into seventeen watertight compartments. Coffer dams 3 feet wide are on each side of the vessel and are packed with corn-pith cellulose. The upper deck, and the main deck outside the superstructure are laid with wood, the others are covered with linoleum.

Hold.—In the hold abaft the engine compartments and forward of the boiler compartments are: Magazines, trimming tanks, store rooms, shell rooms and fresh water tanks.

After Platform.—On the after platform are: Shell rooms, magazines, store rooms, steering engine, handling room for 13-inch turret.

Forward Platform.—On the forward platform are: Shell rooms,

magazines, store rooms, handling room for 13-inch turret, trimming tanks.

Orlop or Splinter Deck.—On the orlop or splinter deck are: Store rooms, magazines, shell rooms, ventilating-blower rooms, ammunition passage, coal bunkers.

Berth Deck.—On this deck are situated the sick bay, berthing space for men, quarters for officers and petty officers, coal bunkers.

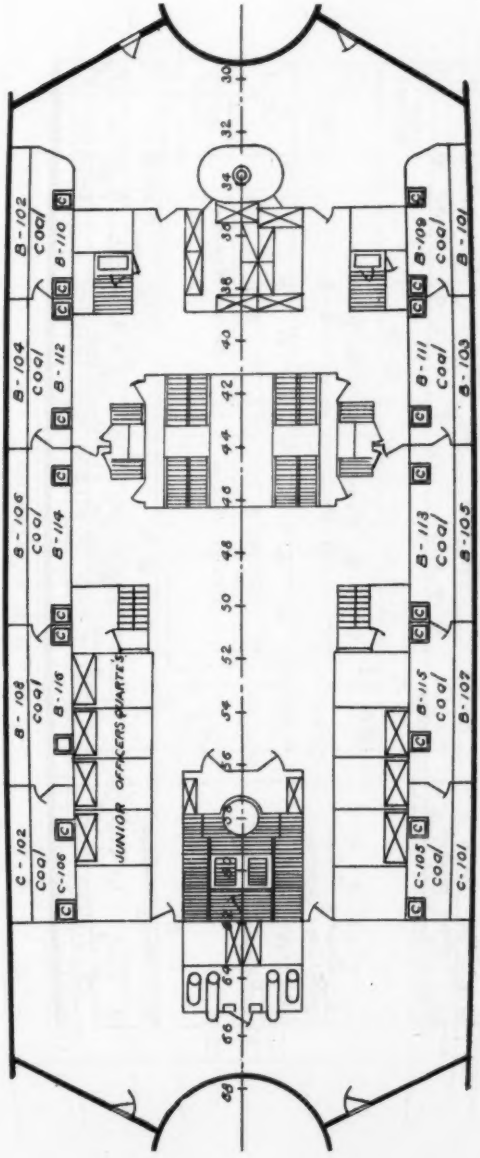
Main Deck.—On this deck are situated the berthing space for men, water closets, after 13-inch turret.

Coal Bunkers.—There are thirty-two coal bunkers with a total capacity of 1,355.2 tons at 43 cubic feet to the ton. There are fourteen in the hold, with a capacity of 960.6 tons; eight on the orlop or splinter deck, with a capacity of 136.8 tons, and ten on the berth deck, with a capacity of 257.8 tons. They are filled through chutes and trunks fitted between decks. The trunks are fitted with shunt doors at the top where they pass through the bunkers on the berth deck, and vertical doors at the bottom, so that the coal can be passed from these bunkers below. Armored shutters are fitted in the trunks where they pass through the protective deck. Trolley rails and buckets are fitted fore and aft in the outboard hold bunkers with portable sections in the wake of watertight doors, for the transportation of coal. Escape doors are fitted to all bunkers; those on the berth deck open to the main deck (circular plates), those on splinter deck to ammunition passage (swinging doors), those in the hold to deck of ammunition passage (circular plates). Steam fire-extinguishing pipes are fitted to all bunkers and holds. Over the watertight sliding doors of hold bunkers are fitted screen doors in the shape of a hood, designed to take the weight of the coal from the watertight doors, so that they may be easily closed at any time.

There are two ash chutes on each side fitted inside the vessel, the upper ends of the chutes extending above the main deck, where hoppers are formed inside, and so arranged that the fire-main pressure may be used to clear the chutes. Trolley ways for carrying the ash buckets are fitted under the deck beams above, leading from the ash hoists to the chutes.

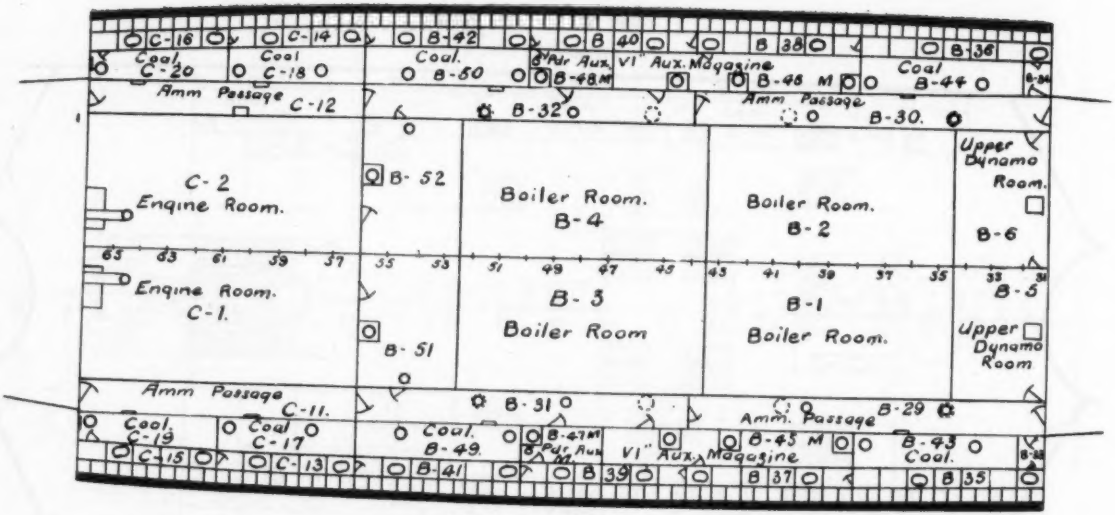
Coal-Bunker Capacities.—Hold compartments:

Compartment.	Cubic feet.	Tons.
B— 7	3,928	91.3
B— 8	3,928	91.3
B— 9	3,697.3	85.9
B—10	3,695.7	85.9
B—11	3,769	87.6
B—12	3,766.5	87.6
B—13	3,702.6	86.1
B—14	3,692.6	85.8
B—15	3,040.4	70.7
B—16	3,104.8	72.2
C— 3	1,421.5	33
C— 4	1,421.5	33
C— 5	1,080.3	25.1
C— 6	1,080.3	25.1
	Total,	960.6
Splinter-deck compartments:		
B—43	811.6	18.9
B—44	811.6	18.9
B—49	863.1	20.1
B—50	863.1	20.1
C—17	678.1	15.7
C—18	678.1	15.7
C—19	592.2	13.7
C—20	592.2	13.7
	Total,	136.8
Berth-deck compartments:		
B—109	1,018.5	23.6
B—110	1,018.5	23.6
B—111	1,070.3	24.9
B—112	1,070.3	24.9
B—113	1,336.9	31
B—114	1,336.9	31
B—115	1,187.8	27.6
B—116	1,187.8	27.6
C—105	941.1	21.8
C—106	941.1	21.8
	Total,	257.8
	Total of all bunkers,	1,355.2



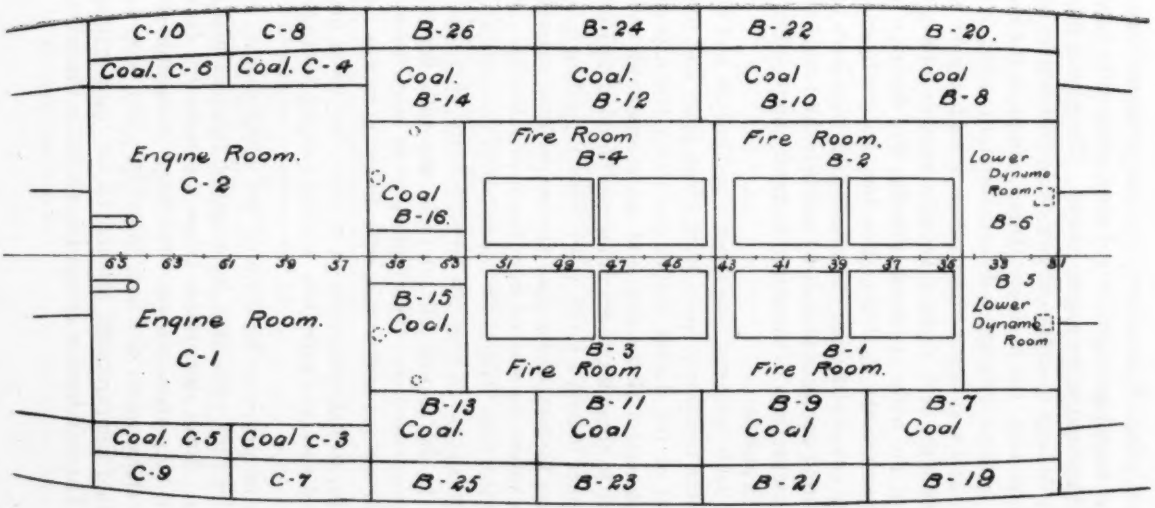
U. S. S. "ALABAMA."

BUNKER PLAN.—BERTH DECK.



U. S. S. "ALABAMA."

BUNKER PLAN.—SPLINTER DECK.



U. S. S. "ALABAMA."

BUNKER PLAN.—HOLD.

Drainage System.—For draining the compartments there is a 14-inch main drain pipe of galvanized steel, which runs from the forward boiler rooms to the engine rooms. This pipe is fitted in order to enable large quantities of water to be removed by means of connections with the main circulating pumps in the engine rooms. It is fitted with screw-down non-return valves in each boiler room and with screw-down valves in each engine room, all worked from the deck above. The suction pipes leading from the valves in the boiler rooms do not dip into drainage wells, but have their mouths located as low down as practicable and covered with coarse strainers. The main drain pipe leads to the main circulating pumps, and is also provided with branches and flanges for connections to the principal steam pumps, other than the main feed pumps.

Two manifolds are fitted, one forward and one aft, and others throughout the machinery space, from which pipes lead to each compartment of the double bottom, trimming tanks, principal lower compartments of the ship, and sea connections. Where pipes leading to these manifolds drain water from compartments above the inner bottom a small well is fitted, projecting 6 inches into the inner bottom, and a plate strainer is fitted around the pipes. The valves are worked from the manifolds only. Manifolds are fitted with flanges for steam-pump connections. A 5-inch drain pipe is run from the forward to the after manifold, connecting to the suction side of each, and with flanges for steam-pump connections. The drain pipe in each boiler and engine room is fitted with a flange for direct connection for the steam pumps in that compartment, in order that water may be drained from such compartments without passing through the manifold.

There is no provision for draining the compartments above the protective deck other than by means of scuppers and portable pumps. Small store rooms below the protective deck are also drained by portable pumps. The larger compartments are drained by sluice valves and with short pipes with stop valves, emptying into compartments reached by the main drainage of the ship. The coal-bunker compartments below the protective

deck are drained by means of the vertical sliding doors of the bunkers. The waste water from the bath and wash rooms is led by pipes to a tank situated in after athwartship coal bunker and is pumped overboard by a steam pump.

The boats carried are as follows:

1 36-foot steam cutter,
1 33-foot steam cutter,
2 33-foot launches,
5 30-foot cutters,
1 30-foot barge,
2 30-foot whale boats,
1 32-foot gig whale boat,
2 22-foot dinghies,
1 balsa.

HULL DATA.

Length between perpendiculars, feet	368
on L.W.L., feet	368
over all, feet and inches	373–11 9/16
Beam, extreme, feet and inches	72– 2½
at L.W.L., feet and inches	72– 2½
Ratio of length to beam	5.1 to 1.
Depth in hold from top of main deck beams to top of floor, feet and inches	32– 4⅛
Height of superstructure above main deck beams, feet and inches	15– 1½
Draught forward, seagoing trim, feet	23.5
aft, seagoing trim, feet	23.5
mean, seagoing trim, feet	23.5
Displacement, seagoing trim (load draught), tons	11,562
per inch at L.W.L., tons	47.9
Area of immersed midship section, square feet	1,610
Center of gravity of L.W.L. plane, aft of midship section (frame No. 46), feet	1.5
Center of buoyancy above bottom of keel, feet	12.95
forward of midship section, feet	1.1
gravity above bottom of keel, feet	29.81
Transverse metacenter above center of buoyancy, feet	17.12
Longitudinal metacenter, above center of buoyancy, feet	372
Coefficient of fineness on extreme dimensions	.637
of midship section	.948
of L.W.L.	.756
Cylindrical coefficient	.683
Number of frames	94
watertight compartments	281

ARMOR.

The armor consists of a side belt of hardened nickel-steel extending 3½ feet above the 23½-foot load line to 4 feet below it. The outside surface conforms to the shape of the vessel, with a thickness of 16½ inches at the lower edge of chamfer at top of armor, from which point it tapers to 9½ inches in thickness at top of armor shelf. The maximum thickness of armor at the water line is maintained from the after end of the belt to the forward coal-bunker bulkhead. From the forward coal-bunker bulkhead to center of forward turret the armor tapers in thickness to 10½ inches, from thence it tapers in 30 feet to 4 inches, which thickness is maintained to the bow of the vessel.

The diagonal armor extends from the slope of the protective deck to top of armor belt at the forward and after ends of machinery and boiler spaces and has a thickness of 12 inches for its entire depth.

The casemate armor above belt has a thickness of 5½ inches. It is worked from the center of forward barbette to the center of the after barbette. The ends of the belt are formed by the diagonal armor 5½ inches thick worked from the sides of the vessel to the barbette armor. The superstructure armor is 5½ inches thick.

The doors through the armor are fitted with hinged armor plates and light plate doors. The armor forward and casemate on upper deck amidship is 6 inches thick.

The barbettes are 15 inches thick at the point and 10 inches thick at the rear. The 13-inch turrets are 14 inches thick at the front and 14 inches thick at the rear.

The forward conning tower is 10 inches in thickness. The armor plates are 7 feet above the top of support and are pierced with sight holes. The tower is fitted with a flat top 2 inches in thickness.

The after conning tower or signal station is 6 inches in thickness, forged in one piece.

A steel tube 7 inches in thickness by 12 inches inside diameter extends from the forward conning tower to the protective deck, to protect the electric wires, voice tubes, connections to engine room, telegraphs, steering gear, and engine tell-tales.

ARMAMENT.

The main battery consists of four 13-inch breech-loading rifles, 35 calibers, mounted in pairs in two balanced turrets, on the fore-and-aft center line of the ship, on the main deck; ten 6-inch rifles on main deck; four 6-inch rifles on upper deck.

The secondary battery consists of

4 6-pounder Hotchkiss guns in cabins,
2 6-pounder Hotchkiss guns on forward main deck,
2 6-pounder semi-automatic Maxims on bridge deck,
8 6-pounder semi automatic Maxims on bridges,
2 3-inch field guns on bridge deck,
2 1-pounder Hotchkiss guns on bridge deck,
4 1-pounder automatic Maxim guns in tops,
4 Colt 6-mm. automatic guns on bridge deck.

There are four tubes for torpedoes, two on each side, discharging directly from the berth deck. The turrets are trained by electricity and the 13-inch guns are elevated and loaded by the same power. The 6-inch guns are worked by hand on central-pivot carriages.

MAIN ENGINES.

There are two vertical, inverted, direct-acting, triple-expansion engines placed abreast of each other in separate watertight compartments, the high-pressure cylinder being forward. Each cylinder is supported by an inverted Y-frame on one side, and by two hollow forged-steel cylindrical columns on the other. The high-pressure cylinders are steam jacketed around the working linings, and the intermediate and low-pressure cylinders are steam jacketed around the working linings and at both ends. There are no starting valves on the cylinders, provision being made for starting the engines by admitting live steam to the receivers. The main valves are of the single-ported piston type, there being for each engine, one for the H.P., two for the I.P., and four for the L.P. cylinder. The main valves are provided with balance pistons, the cylinders of which form part of the upper covers of the valve chests. The valve gear is of the Stephenson link type with double-bar links. There are no independent cut-off valves, but provision is made to cut off in each

cylinder varying from .5 to .7 of the stroke, by means of a block to which the suspension links are attached, which block can be moved by hand-screw gear in a slot in the end of the arm on the reversing shaft.

The main pistons are steel castings, dished, and are each fitted with two packing rings $\frac{5}{8}$ inch wide and $\frac{3}{4}$ inch thick. The piston rods and connecting rods are of forged steel. The crossheads are of forged steel, and each has a manganese-bronze slipper, the sliding faces of which are fitted with white metal. The crosshead guide to take the thrust when going ahead is of cast iron bolted to the inverted Y frame. The back of the guide is recessed for the circulation of water. Cast-iron lips are bolted on each side of the ahead guide to take the thrust when backing. The eccentrics are of cast iron. Each backing eccentric is securely keyed on the shaft, and each go-ahead eccentric is secured to the corresponding backing one by through bolts in slotted holes. The eccentric straps are of composition faced with white metal, and the eccentric rods of forged steel. The I.P. and L.P. valve stems have manganese-bronze crossheads which take hold of the link blocks directly. Each engine bed plate consists of three sections of steel castings. The reversing gear consists of a steam cylinder and a hydraulic controlling cylinder acting directly on two arms fixed on the reversing shaft. The steam piston rod is secured to a steel crosshead connecting with the arms on the reversing shaft. The piston rod passes through the controlling cylinder. There is a by-pass valve on the controlling cylinder which is worked by a continuation of the stem of the steam piston valve. These valves are worked by a system of differential levers, the primary motion being derived from the hand lever on the working platform, and the secondary motion from a pin on the reversing arm. There is a stop cock in the by-pass pipe of hydraulic cylinder, and a pump for reversing by hand is connected to the hydraulic cylinder.

There is a double-cylinder, simple, inverted, turning engine, with cylinders 7 inches in diameter and 7 inches stroke, secured to the high-pressure Y frame. It drives a worm wheel in the forward end of the high-pressure crank shaft by means of worm

gearing and a second worm; the latter is made to slide on a feather key, and is held in place by a collar below and a removable key above it. A ratchet is fixed to the shaft of the engine for turning by hand. Each engine is fitted with a disc stop valve 13 inches in diameter, having a screw stem and a balance piston and a butterfly throttle. The main steam pipes of and above 9 inches in diameter are banded with $\frac{1}{8}$-inch steel bands 2 inches wide, spaced 6 inches between centers.

In each engine room there is a distributing oil tank supplied from the main oil tanks by a steam pump in the port engine room. The tanks are fitted with overflow pipes. Distributing pipes lead to the various manifolds, which are fitted with adjusting valves, and from the manifolds tubes lead to the various parts to be lubricated. The suction pipe of the pump is fitted with a strainer.

ENGINE DATA.

Cylinders, number for each engine	3
diameter of H.P., inches	33½
I.P., inches	51
L.P., inches	78
Stroke of pistons, inches	48
Valves, diameter of H.P. (one for each engine), inches	14
I.P. (two for each engine), inches	20¾
L.P. (four for each engine), inches	18
Balance pistons, diameter of H.P., inches	8
I.P., inches	8
L.P., inches	7½
Valve stems, H.P. (1), diameter, inches	2¾
through valve, inches	1½
I.P. (2), diameter, inches	2¾
through valve, inches	1¾
L.P. (4), diameter, inches	2¾
through valve, inches	1¾
Main steam pipe, diameter, inches	13
Exhaust pipe to I.P. cylinder, diameter, inches	15½
L.P. cylinder (4), diameter, inches	15
condenser (2), diameter, inches	20½
Volume swept by H.P. piston, per stroke, cubic feet	23.87
I.P. piston, per stroke, cubic feet	56.13
L.P. piston, per stroke, cubic feet	132.12
Ratio of net area of H.P. to I.P. pistons	2.35
I.P. to L.P. pistons	2.35

Ratio of net area of H.P. to L.P. pistons	5.53
Clearance of H.P. cylinder, per cent. top, 20.06; bottom,	17.02
I.P. cylinder, per cent. top, 20.71; bottom,	18.12
L.P. cylinder, per cent. top, 15.59; bottom,	20.28
Piston rods, diameter, inches	7½
length from piston to crosshead, feet and inches	7–2
axial hole, diameter, inches	3½
Connecting rods, length from center to center, feet	8
diameter of upper end, inches	7⅞
lower end, inches	8¼
axial hole, diameter, inches	4½
crosshead bolts (2), diameter, inches	4¼
crank-pin bolts (2), diameter, inches	4¼
crosshead pin, length, inches	14
diameter, inches	9¾
axial hole, diameter, inches	4
Crossheads, surface (ahead), square inches	459
(backing), square inches	357
Reversing gear, steam cylinder, diameter, inches	14
stroke, inches	20

Shafting and Bearings.—The crank, line and propeller shafting is hollow and of forged steel. The crank shaft of each engine is in three interchangeable sections, the cranks placed at angles of 120 degrees, and for the ahead motion follow in the order H.P., I.P. and L.P. There is one section of line shafting supported on two spring bearings. The propeller shafts are covered with a composition casing from the forward coupling to 1 inch inside of propeller hub; they extend far enough inboard to allow the inboard stern bearings to be withdrawn without moving the shafts. The inboard coupling of the propeller shafts consists of a forged-steel collar 15 inches long secured by three steel keys. Back of this collar another collar is fitted in halves, of the same diameter, and secured to it and to the coupling on the line shaft by six 3½-inch fitted-steel bolts.

Each thrust bearing is of cast iron of the horse-shoe pattern. The end and side walls of the pedestal form an oil trough inside of which, both forward and aft the horse shoes, is a composition bearing lined with white metal for taking the weight of the shaft. The cap of this bearing is of cast iron lined with white metal. There is an oil cup in the top of each horse shoe,

from which there is an oil hole to each collar, the white metal being channeled for the distribution of the oil. At each end of the bearing there is a divided stuffing box and gland to prevent the escape of oil. The pedestal is bolted to a cast-iron sole plate fitted with wrought-iron wedges at each end of the pedestal for adjusting the bearing fore-and-aft.

Crank shafts, diameter, inches	14¾
axial hole, diameter, inches	7½
coupling disc, diameter, inches	27½
thickness, inches	3½
tapered coupling bolts in one flange (6), diameter, inches	3⅛ to 3¾
journals, diameter, inches	14¾
length, inches	16½
length of each section, feet and inches	8–10
Crank pins, diameter, inches	14¾
length, inches	17
axial hole, diameter, inches	7½
Crank webs, width, inches	16½
thickness, inches	10
Thrust shafts, diameter, inches	14
of coupling discs, inches	27½
axial hole, diameter, inches	9
collars, number, each shaft	11
diameter, inches	21½
thickness, inches	2
distance between, inches	3½
surface, total for both engines, square inches	4,600.48
length, feet and inches	12–6
Line shafts, diameter, inches	14
axial hole, diameter, inches	9
diameter of coupling discs, inches	27½
total length, feet and inches	36–6
Propeller shafts, diameter, inches	14¾
axial hole, diameter, inches	9¾
length, feet and inches	52–4⅜
Stern bracket bearing, length, inches	43¾
diameter, inches	16¾
Stern-tube bearing, length, forward, inches	33
aft, inches	36

Main Condensers.—There are two main condensers, one for each engine. The shells are of wrought-steel plate, stiffened with tees, butted with a single-butt strap on the inside, and with angle iron

riveted around each end to form flanges for securing the water chests. The water chests and the tube sheets are of composition. The circulating water passes through the tubes. Baffle plates are fitted to direct the steam over the tubes, and plates are fitted for supporting the tubes and to act as baffle plates. The water chest for the entrance and exit of circulating water has a division plate fitted with a valve worked by a lever on the outside of the condenser to allow the circulating water to pass overboard direct when the valve is open.

Diameter of shell (inside), feet and inches	5–10½
Thickness of shell, inch	$\frac{5}{16}$
Length over heads, feet and inches	13– 8$\frac{7}{8}$
Tubes, diameter (outside), inch	$\frac{5}{8}$
length between tube sheets, feet and inches	11– 6
thickness, No. 18 B.W.G., inch	.049
number in each condenser	3,722
Cooling surface, each condenser, square feet	7,002
total, square feet	14,004
Ratio of total cooling to total heating surface	1.55

Main Air Pumps.—For each engine there is a Worthington twin-cylinder, single-acting air pump. The steam cylinders are placed directly over the pump cylinders, the pump rods and piston rods forming a continuous length. The pumps are connected by means of a beam which is pivoted at its center, and from which beam the valve motion is actuated. The beam receives its motion through links swinging from crossheads on the pump and piston rods. The pump valves are each made of three flat discs of rolled manganese-bronze.

Diameter of steam cylinders (2), inches	12
pump cylinders (2), inches	25
Stroke, inches	18
Diameter of piston rods, inches	2$\frac{7}{16}$
pump rods, inches	2¾
Kind and diameter of pump valves (manganese-bronze), inches	4$\frac{1}{8}$
Ratio of volume swept by L.P. piston per stroke to that of the two air pump buckets per stroke	13.08 to 1

Main Circulating Pumps.—For each main condenser there is a centrifugal, double-inlet circulating pump, which is arranged

to draw from the sea, bilge and main drain pipe, and to discharge into the condenser. It is driven by an independent, vertical, single-cylinder engine. Each pump is capable of discharging 10,000 gallons of water per minute from the bilge. The sea and bilge injection valves are fitted with a self-locking arrangement so that both cannot be opened at the same time.

Diameter of steam cylinder, inches	14
Stroke, inches	12
Diameter of pump runner, inches	54
Width of runner, inches	at hub, $7\frac{3}{8}$; at rim, $3\frac{1}{2}$
Diameter of inlet nozzle, inches	$15\frac{1}{2}$
outlet nozzle, inches	$15\frac{1}{2}$

Auxiliary Condensers.—There is in each engine room an auxiliary condenser connected with the auxiliary exhaust pipes. Each has a horizontal combined air and circulating pump, the steam cylinder being between the water cylinders and all the pistons being on one rod. The tubes are arranged and packed as in the main condensers.

Cooling surface, one condenser, square feet	800
Diameter of steam cylinder, air and circulating pump, inches	10
water cylinder, inches	12
air-pump cylinder, inches	12
Stroke, common, inches	12

Feed-Water Heaters.—In each engine room there is a feed-water heater which is similar in construction to a condenser. The heaters are located between the main feed pumps and the feed main, and the valves are so arranged that the feed water can pass directly to the feed main without passing through the heaters.

Screw Propellers.—The propellers are of manganese-bronze, each with three adjustable blades. They are true screws, with pitch variable from 16 feet 6 inches to 18 feet 6 inches. Each boss is secured to the shaft by a feather key and a wrought-steel nut screwed on and locked in place. The end of the hub is covered with a composition cap.

Number of blades	3
Diameter, feet	16
of hub, feet and inches	3–$11\frac{1}{2}$

Length of hub, feet and inches	3- 1
Pitch as set, feet	17½
Greatest width of blade (4½ feet from axis), inches	58
Helicoidal area of each screw, square feet	76
Projected area, square feet	62
Disc area, square feet	201.06
Pitch ÷ diameter	1.09
Helicoidal area ÷ disc area	.376
Projected area ÷ disc area	.308
Disc area of both screws ÷ I.M.S.	.249
Immersion of center at mean draught (23½ feet), feet	14¾
Center above lowest point of keel, feet	8¾
from center line of vessel, feet and inches	13-0¾

BOILERS.

There are eight single-ended boilers of the horizontal, return, fire-tube type, all of steel. The shells are of one course of four plates. The longitudinal joints of the shells are treble riveted with double-butt straps. The joints in furnaces and combustion chambers are single riveted. There are two combustion chambers in each boiler, and are so arranged that there is one combustion chamber for each wing and its adjacent furnace. The tops of the combustion chambers are braced by girders. The combustion chambers are stayed to each other and to the shell by screw stays screwed into both plates and fitted with nuts, the nuts being set up on beveled washers where stays do not come square with the plates. Each furnace is in one piece and corrugated, riveted to flanges of front head and flanged and riveted to combustion chamber plates. The grate bars are of cast iron, and those at the sides are made to fit the corrugations. The bridge walls are of cast iron, and extend to the back of the combustion chambers. Each boiler has circulating plates fitted at each side of each nest of tubes. Each boiler is bolted to angle irons which are riveted to the saddles. The boilers are bolted rigidly to the angle iron of outboard saddles, but the inboard angle irons have oval holes to allow for expansion. Each boiler has two 4¼-inch spring safety valves placed on the stop-valve nozzles, the two valves being in one case. The internal feed pipes are fitted with suction branches to the bottom of the boilers, the combined feed and circulating water being dis-

charged near the water level through down-pointing branches. The auxiliary feed pumps may be used to circulate the water. The tubes in the boilers are of charcoal iron. The boilers are placed athwartships in four watertight compartments. There are two smoke pipes, one for the starboard boilers and one for the port boilers, the pipes being abreast each other.

BOILER DATA.

Steam pressure designed, pounds	180
Number of boilers	8
Length, feet and inches	9–11½
Diameter, outside, feet and inches	15– 8
Thickness of shell, inches	1 7/16
heads, top, inches	1 5/8
bottom, inch	¾
tube sheets, inch	¾
furnace, inch	9/16
Combustion chambers, number in each boiler	2
thickness of plates, inch	9/16
width at top, feet and inches	2– 3 7/16
depth, feet and inches	9–11½
Furnaces, greatest external diameter, feet and inches	3– 7
least internal diameter, feet and inches	3– 3
length of grate, feet and inches	6– 8½
number in each boiler (corrugated)	4
Tubes, outside diameter, inches	2½
length between tube sheets, feet and inches	6– 6¼
number of ordinary, each boiler	388
number of stay, each boiler	150
spaced vertically, inches	3½
horizontally, inches	3¾
thickness of ordinary, No. 9, B.W.G., inch	.148
stay, No. 6, B.W.G., inch	.203
Diameter of rivets in shell sheets, inches	1 5/16
screw stays, inches	1½
Number and diameter of through upper braces, inches	twenty-two of 2½; two of 2
lower braces, inches	three of 1¾
braces around lower manholes, inches	nine of 1¾
from head to back tube sheets, inches	eight of 2
Heating surface, tube, each boiler, square feet	2,296.12
furnace, each boiler, square feet	141.93
combustion chambers, each boiler, square feet	273.39
total, each boiler, square feet	2,711.44
Grate surface, each boiler, square feet	87.2

Area through tubes, each boiler, square feet	13.87
over bridge walls, each boiler, square feet	7.68
Smoke pipes (2), diameter (elliptical), feet and inches	10–9; 6–3
area of both, square feet	115.36
height above lowest grates, feet	86½
Number and diameter of safety valves (2 on one base), inches	4½
Diameter of boiler main stop valves, inches	6½
auxiliary stop valves, inches	6
Totals for eight boilers:	
Heating surface, tube, square feet	18,368.96
furnace, square feet	1,135.44
combustion chambers, square feet	2,187.12
total, square feet	21,691.52
Grate surface, square feet	697.6
Area through tubes, square feet	110.96
over bridge walls, square feet	57.41
Volume of steam room, cubic feet	3,224
Area of water surface, square feet	1,051.2
Ratios:	
Tube H.S. to G.S.	26.83 to 1
Furnace H.S. to G.S.	1.66 to 1
Combustion chamber H.S. to G.S.	3.19 to 1
Total H.S. to G.S.	31.68 to 1
Area through tubes to G.S.	.162 to 1
Steam room per square foot of G.S.	37.67 to 1
Volume of furnaces and combustion chambers above grates to G.S.	3.171 to 1

Forced Draft.—The closed fire-room system of forced draft is used. The air is supplied by eight Sturtevant blowers, two in each fire room. The fans are driven by two-cylinder vertical simple enclosed engines with cranks at 180 degrees.

Diameter of cylinders, inches	5
Stroke, inches	4
Diameter of fan, inches	60
Width of fan at rim, inches	14
Area of induction nozzle, square inches	1,060.7
eduction nozzle, square inches	1,380

Feed Pumps.—There are two main and four auxiliary feed pumps of the Admiralty type, Worthington, duplex, and all of the same size, 10 inches by 7 inches by 12 inches, each capable of delivering 400 gallons per minute. The main feed pumps are located one in each engine room, and draw from the feed tanks and air pump suctions and deliver to the boilers. The auxiliary

feed pumps are situated in the fire rooms, and draw from the feed tanks, air-pump suctions, sea, drainage system and boilers, and discharge into the boilers, fire main and overboard. The discharge of the main and auxiliary pumps are independent; the main feed pumps discharging only to the boilers.

Fire and Bilge Pumps.—In each engine room there is a vertical, duplex, Worthington pump which draws from the sea, bilge and drainage system and delivers into the fire main and overboard. They have a capacity of 400 gallons per minute. The steam cylinders are 10 and the water cylinders 7 inches in diameter; stroke, 12 inches.

Water-Service Pumps.—In each engine room there is a vertical, duplex, Worthington pump which draws from the sea and delivers into the water-service pipes, the distiller circulating pipes and fire main. They have a capacity of 400 gallons per minute. The steam cylinders are 10 and water cylinders 7 inches in diameter; stroke, 12 inches.

Bilge and Sanitary Pumps.—There are two single-acting plunger pumps on each main-engine shaft driven by eccentrics; one pump on each shaft is arranged to draw from the bilge and discharge overboard, the other to draw from the sea and discharge into the flushing main.

Feed Tanks.—There is a feed tank of 2,000 gallons capacity in each engine room. A part of the tank is fitted as a filter into which the water from the air pumps is delivered, the water descends through the filtering material and discharges into the main portion of the tank. Each tank has a man-hole with bolted cover, glass water gage, shut-off cocks and drain cocks. Each tank and filter has the following pipe connections:

A discharge pipe from the air pump;

An overflow pipe to the bilge, so arranged that any water passing down it may be seen;

A suction pipe to feed pumps, with valve;

Drain pipes from traps;

A pipe and valve for drawing water from filter box;

A vapor pipe.

Each feed-pump suction is provided with a balanced valve

operated by a copper float in the feed tank, so arranged that it will allow no air to enter the feed pipes.

Steam Traps.—The separators, jackets, main and auxiliary steam pipes, the radiators, and all places where condensed steam can accumulate are fitted with drain pipes and cocks, or valves, and with automatic traps which discharge into the feed tanks. The traps are provided with by-pass pipes and valves, for convenience in overhauling.

Grease Extractors.—A grease extractor is fitted on the discharge of both main feed pumps. It is similar to a Macomb strainer: the basket is perforated and covered with burlap through which the water filters. By-pass pipes and valves are fitted to feed the boilers while overhauling the extractor.

Engineers' Workshop.—The workshop is fitted on the splinter deck, forward of the engine rooms. The shafting is driven by a single vertical engine 8 inches by 8 inches. It is provided with the following tools:

1 24-inch screw-cutting lathe,
1 14-inch screw-cutting lathe,
1 column-shaping machine,
1 double-geared drilling machine,
1 combined hand punch and shears,
1 emery grinder.

Ash Hoists.—In each fire-room hatch there is a Williamson double-reversible ash hoist, by means of which (with a steam pressure of 60 pounds) one bucket of ashes of 300 pounds can be hoisted in 5 seconds. The steam cylinders are $4\frac{1}{2}$ inches in diameter with a stroke of $4\frac{1}{2}$ inches.

Distilling Apparatus.—In the engine rooms are two Williamson Bros. straight-tube distillers, and two evaporators, Bureau's pattern, with a combined capacity of potable water at a temperature of 90 degrees Fahrenheit of 7,500 gallons in twenty-four hours. The water-service pumps are used for the distiller circulating water. The distillers are arranged so that they can take steam from the auxiliary steam pipe in case of emergency. A filter and water meter are fitted to each distiller. There are two evaporator feed pumps of the Worthington horizontal duplex

type, two combined brine and fresh-water pumps of the same type, also two trap pumps.

In addition to the above there is an evaporating and distilling plant, situated in the drying room on the main deck, capable of distilling 5,000 gallons of water at a temperature of 90 degrees Fahrenheit in twenty-four hours.

Ice Machines.—There are two Allen dense-air ice machines capable of producing the cooling effect of two tons of ice per day. The cooling pipes connect to the ice tanks, refrigerating rooms and scuttle butt. Steam cylinder, 7 inches diameter; air-compressor cylinder, 5¾ inches diameter; air-expansion cylinder, 4¾ inches diameter; stroke, 10 inches.

Ventilating Fans.—In each engine-room hatch and in the upper part of each engine room there is a ventilating fan for engine-room ventilation only. For ventilating the ship generally there are six fans situated on the splinter deck; they are so arranged that by means of reversible valves they can draw from or force air to the various parts of the ship. In the steering-engine room there is an electrical exhauster driven by a 3-H.P. electric motor. There are two electrical fans in dynamo rooms each driven by a 2-H.P. motor. The fans were furnished by the Sturtevant Company.

Diameter of steam cylinders, inches	4
Stroke, inches	3
Diameter of fan, inches	50
Width of fan, inches	17½

Steering Engine.—On the after platform deck there is a Williamson Bros. combined hand and steam-steering engine, cylinder, 14 inches in diameter; stroke, 11 inches. The engine is connected to a horizontal fore-and-aft shaft, on the after end of which is a right-and-left-handed screw which works in the crossheads, one on the starboard and one on the port side, the rudder yoke being connected by rods to these crossheads. With the steam gear the rudder can be put from hard over to hard over in 20 seconds when the ship is going full speed. The engine is controlled from the conning tower and pilot house by means of hydraulic telemotor or by wire rope.

Steam Windlass.—There is a steam windlass, built by "Hyde Ship Windlass Company," situated forward on the main deck. The engine is a reversible, double-cylinder, vertical, direct-acting, with cylinders 15 inches in diameter and a stroke of 14 inches. The windlass is fitted with two wild cats to take the chains.

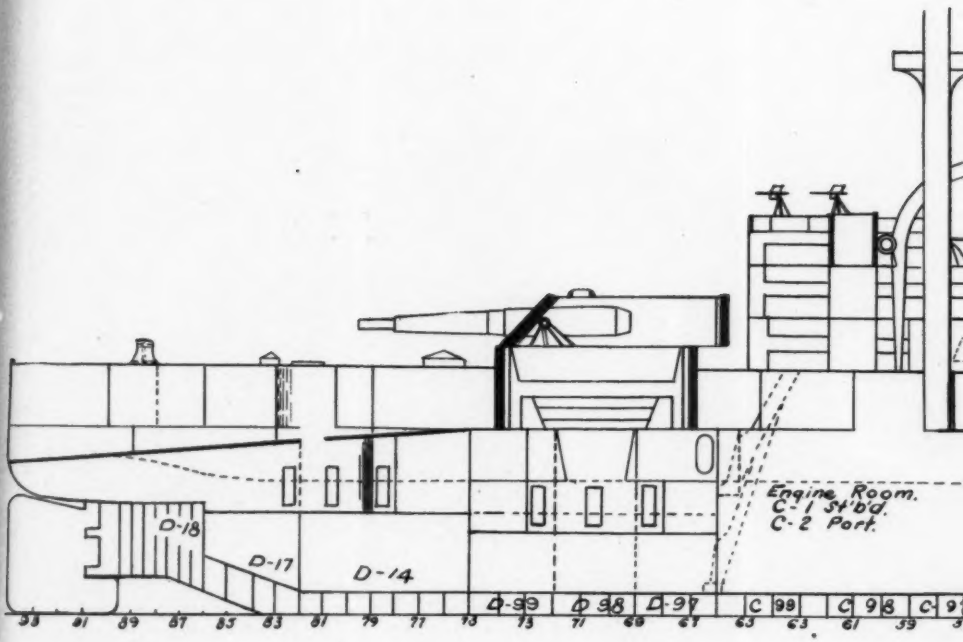
Steam Winches.—There are four steam winches situated on the decks, fitted for manipulating wire cables, hoisting coal, and general deck purposes. They were built by Williamson Bros. Cylinders, 8 inches in diameter; stroke, 8 inches.

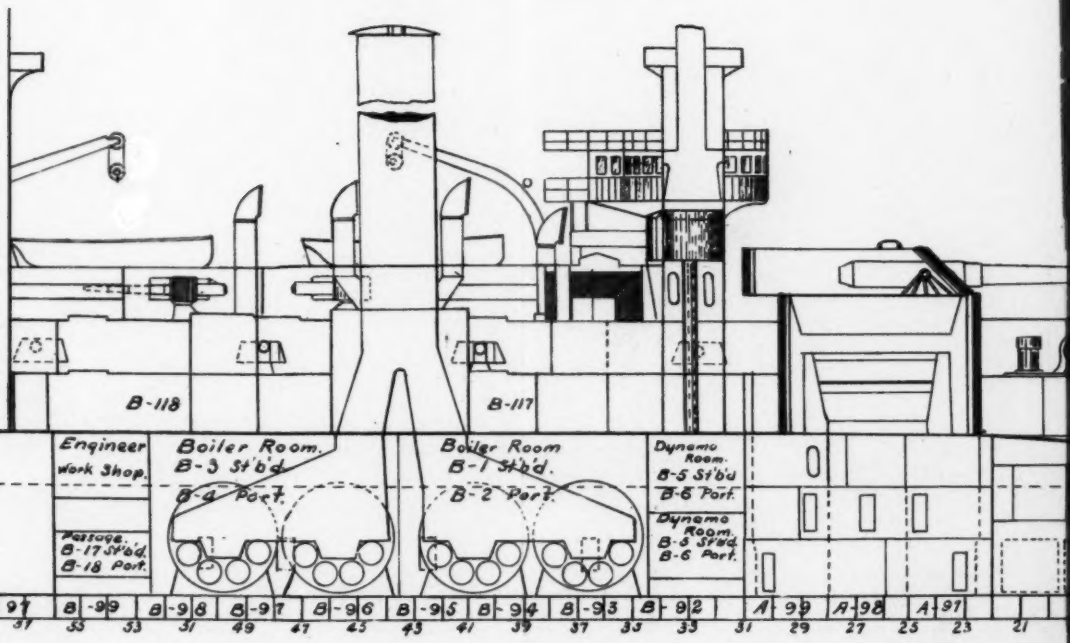
Boat-Crane Engines.—There are four boat-crane engines situated on the upper deck for hoisting in and out boats. They were built by the Williamson Bros. Steam cylinders, 10 inches in diameter; stroke 8 inches.

Air Compressors.—There are two air compressors (Kaselooski system) for torpedoes, situated in the engineers' work shop. Each compressor has one steam cylinder, diameter, 9.252 inches; stroke, 6.299 inches. The compressors are capable of making 275 revolutions per minute with 60 pounds of steam, the air pressure in the accumulator being 1,700 pounds to the inch. The capacity per hour is 19.06 cubic feet.

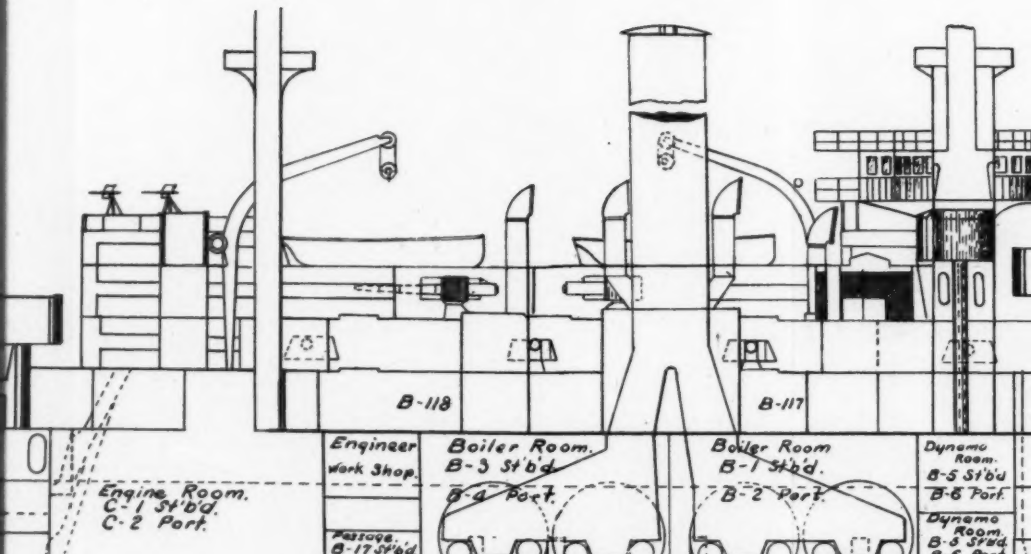
Telegraphs and Revolution Indicators.—There is a Cory mechanical telegraph in each engine room connected to transmitters in conning tower, pilot house and bridge, and a mechanical gong in each engine room with bell pulls on deck. Mechanical tell tales are fitted on the bridge and in the conning tower to show the direction of the revolution of the main engines. In each engine room there is a mechanical indicator to show the relative speed and direction of revolution of the main engines. The two indicators are connected, and each has two hands, a red one which turns with the port engine and a green one which turns with the starboard engine. The hands can be turned on their shafts like the hands of a watch, so that by setting them together the relative speeds of the propellers can be seen as the hands separate or remain together in their revolutions over the dial of the indicator.

Telephones and Voice Tubes.—For communication between the


D-18
D-17
D-14
Engine Room.
C-1 St'b'd.
C-2 Port.

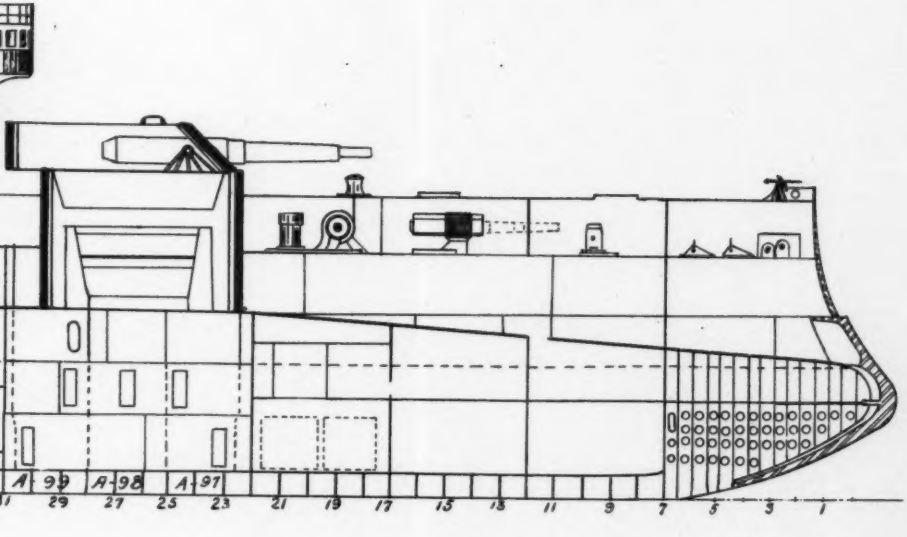

B-118
B-117
Engineer
Work Shop.
Boiler Room.
B-3 St'b'd.
B-4 Port.
Boiler Room
B-1 St'b'd.
B-2 Port.
Dynamo
Room.
B-5 St'b'd
B-6 Port.
Dynamo
Room.
B-5 St'b'd.
B-6 Port.
Passage.
B-17 St'b'd.
B-18 Port.
B-99
B-98
B-97
B-96
B-95
B-94
B-93
B-92
A-99
A-98
A-97

U. S. S. "ALABAMA."


B-118
B-117
Engineer Work Shop.
Boiler Room. B-3 St'bd. B-4 Port.
Boiler Room B-1 St'bd. B-2 Port.
Dynamo Room. B-5 St'bd B-6 Port.
Engine Room. C-1 St'bd. C-2 Port.
Passage. B-17 St'bd. B-18 Port.
Dynamo Room. B-5 St'bd. B-6 Port.
97 C 99 C 98 C 97 B-99 B-98 B-97 B-96 B-95 B-94 B-93 B-92
61 63 63 61 59 57 55 53 51 49 47 45 43 41 39 37 35 33 31

U. S. S. "ALABAMA."

PROFILE INBOARD



various parts of the ship there is a complete system of telephones and voice pipes with call bells. The important stations are connected directly, but for general purposes, in order to lessen the number of pipes, a central station is used.

Steam Cutters.—There are two steam cutters, one 36 feet and one 33 feet long. They have Towne boilers, compound engines and keel condensers. The cylinders for the 36-foot cutter are 4 inches and 8 inches in diameter by 6 inches stroke. Those for the 33-foot cutter are 4 inches and 8 inches in diameter by 6 inches stroke.

Electric Plant.—The installation consists in general of eight 32-kilowatt generating sets, 770 incandescent lights, four searchlights, two sets of signalling apparatus, one main switch board, two auxiliary distributing boards in turrets, together with the necessary wiring, wiring accessories, molding and fixtures and motors. The eight generating sets were made by the General Electric Company. In each set there is a 32-kilowatt dynamo of Multipolar type, having a capacity of 400 ampères at 80 volts, compound wound, six poles. The engines driving the dynamos are compound, tandem, vertical, inverted; diameter of cylinders 9 and 15 inches; stroke, 6 inches. The engines are designed to run at full load at 400 revolutions per minute with 100 pounds of steam pressure. The bed plate is common to both engine and dynamo. The incandescent lights are 5, 16, 32 and 150-candlepower, designed for 80 volts.

Each dynamo is connected to the main switch board and to the equalizer board, the main switch board containing only the panels for the dynamos and lighting and motor bus bars.

The ship is wired on the two-wire system with 80 volts. There are four searchlights, made by the General Electric Company, two mounted on a platform on the main mast and two on forward bridge. The projectors are 50 inches in diameter and hand controlled. All lamps are of the horizontal type with combination hand or automatic feed, and designed for a current of 70 to 80 ampères at about 80 volts. The carbon holders are adjustable, and the life of the carbon is six hours.

The signal apparatus is in two sets, one for each mast, and

consists of four double signal lanterns to each set, cable connections, and key board on forward bridge and in signal tower.

Electric speed indicators and transmitters are fitted in pilot house and conning tower, also electric helm indicators and electric helm telegraphs, and electric main-engine telegraph.

There are thermostats in each coal bunker and in certain store rooms with annunciators on bulkhead forward of cabin.

ELECTRIC MOTORS.

Name.	Number.	Horsepower.	Revolutions.	Volts.	Ampères.
Ventilating fans	1 2	3 2	500	80	32
Small exhausters	2	½	1,000	80	32
13-inch ammunition hoist	4	20	350	80	225
13-inch elevating	4	3½	400	80	45
13-inch rammer	4	5	750	80	55
13-inch turret-turning motors	4	50	400	80	400
6-inch ammunition hoist	14	4.4	400 to 800	80	50
6-pounder ammunition hoist	4	4.4	400 to 800	80	50
3-inch fixed gun ammunition hoist	1	3½	200	80	45
3 hatch lifting	3	1	900	80	11.6

OFFICIAL SPEED TRIAL.

The official speed trial took place off the New England coast on August 28th, 1900. It consisted of two runs over a measured course of 33 nautical miles. The average speed of the two runs was 17.013 knots per hour. The weather was good, the sea smooth, the wind light. All boilers were in use under forced draft. All the steam generated was used by the main engines and auxiliaries except that used in the steering engine and in the engine for the dynamo to light the engine and boiler compartments. The coal used was hand-picked Pocahontas. The boilers steamed freely and steadily. The engines and auxiliaries worked well.

After the trial, the boilers, main engines and auxiliaries were examined and found in good condition.

PERFORMANCE—SYNOPSIS OF STEAM LOG.

Main Engines.

Steam pressure at boilers, per gage, pounds	176.6	
	Starboard.	*Port.*
H.P. steam chest, per gage, pounds	173.1	173.9
first receiver, absolute, pounds	80.7	78.1
second receiver, absolute, pounds	29.	28.8
Vacuum in condensers, inches of mercury	23.68	22.7
Mean effective pressures in cylinders of main engines, pounds per square inch, H.P.	69.75	69.93
I.P.	32.52	32.7
L.P.	15.99	14.86
Mean pressure on L.P. piston, equivalent to aggregate M.E.P. on all pistons, pounds per square inch	42.405	41.39
Pumps, air, main	13.23	
circulating, main	67.76	

Revolutions per Minute.

Main engines	114.03	115.27
Pumps, air	25.45	20.46
circulating	136.45	155.57
feed, double strokes per minute	35.45	
Blower engines, mean revolutions	480	
Speed of ship in knots per hour	17.013	
Slip of propeller in per cent. of its own speed, based on mean pitch	13.61	14.54
Air pressure in fire rooms, in inches of water	1.05	

Indicated Horsepower.

Main engines, H.P.	1,656.5	1,678.7
I.P.	1,816.6	1,846.71
L.P.	2,102.4	1,972.10
Air pump	8.014	5.22
Circulating pump	26.92	40.84
Feed pumps	53.44	
Water-service pumps and all auxiliary machinery in use,	5.6	
Ventilating engines, for engine rooms	10	
Blower engines for forced draft in fire rooms	108	
Dynamo engines	34.8	
Collective of all main engines	11,073	
main engines, air, circulating and feed pumps	11,207 (*A*)	
and auxiliary engines in operation during trial	11,365.9 (*B*)	
Main engines, air, circulating and feed pumps, per sq. ft. G.S.	16.37	
H.S.	.5166	
Cooling surface, square feet, per I.H.P.	1.23	
Heating surface, square feet, per I.H.P.	1.9	

Coal.

Kind and quality.....Pocahontas, excellent.

TEST OF ONE OF THE BABCOCK & WILCOX BOILERS FOR THE *CINCINNATI.*

In the Annual Report of the Chief of Bureau of Steam Engineering there is published a report of a test made on one of the new boilers built by the Babcock & Wilcox Company for the *Cincinnati*, by a board composed of Lieutenant Commander A. B. Willits and Lieutenant B. C. Bryan, U. S. Navy. This test was made June 15 to 22, 1900, at the works of the builders, Elizabethport, N. J., and the following synopsis includes all but the detailed tabulations from which the important data given was deduced.

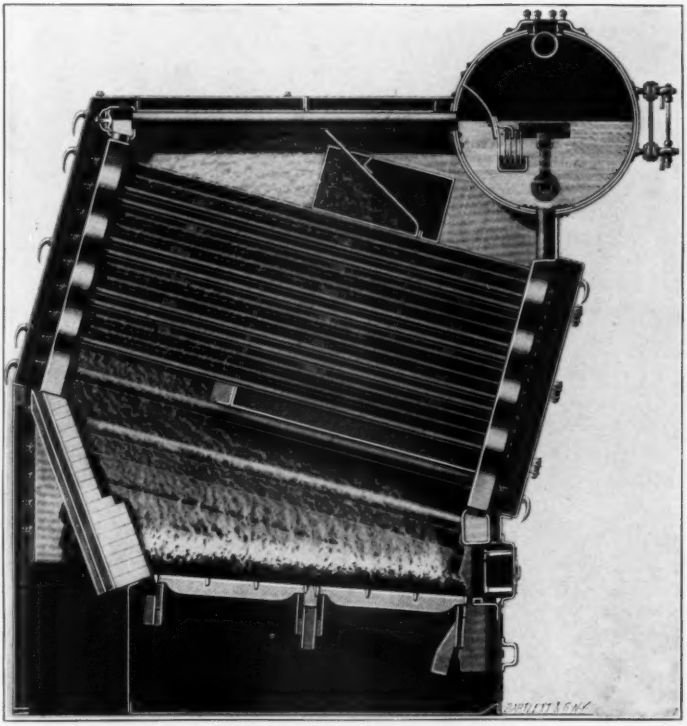
DESCRIPTION OF BOILER AND APPURTENANCES.

The boiler is of the Babcock & Wilcox new marine type, composed entirely of wrought steel, the point of difference between it and the older type of this make of boiler being in the arrangement of baffle plates (as shown in the accompanying sectional view), which compel the products of combustion to pass three times across the tubes before entering the uptake. The small tubes are 2 inches outside diameter, while the bottom tube in each section or element is 4 inches outside diameter. The total heating surface is 2,640 square feet.

The grate is an undivided area of 63.25 square feet, being 6 feet 6½ inches deep and 9 feet 5½ inches wide. It is fired through four properly spaced doors.

BOILER DATA.

Type of boiler, Babcock & Wilcox.
Diameter of top drum, 42 inches (inside).
Length of top drum, 12 feet.
Tubes: Number, 526; 2 inches outside diameter; length, 8 feet. Also 40; 4 inches outside diameter; length, 8 feet 5¼ inches.
Grate surface: Length, 6 feet 8¼ inches; width, 9 feet 5½ inches; area, 63.25.
Grate surface reduced to 5 feet 6 inches length, 52 square feet area, in tests Nos. 5 and 6.



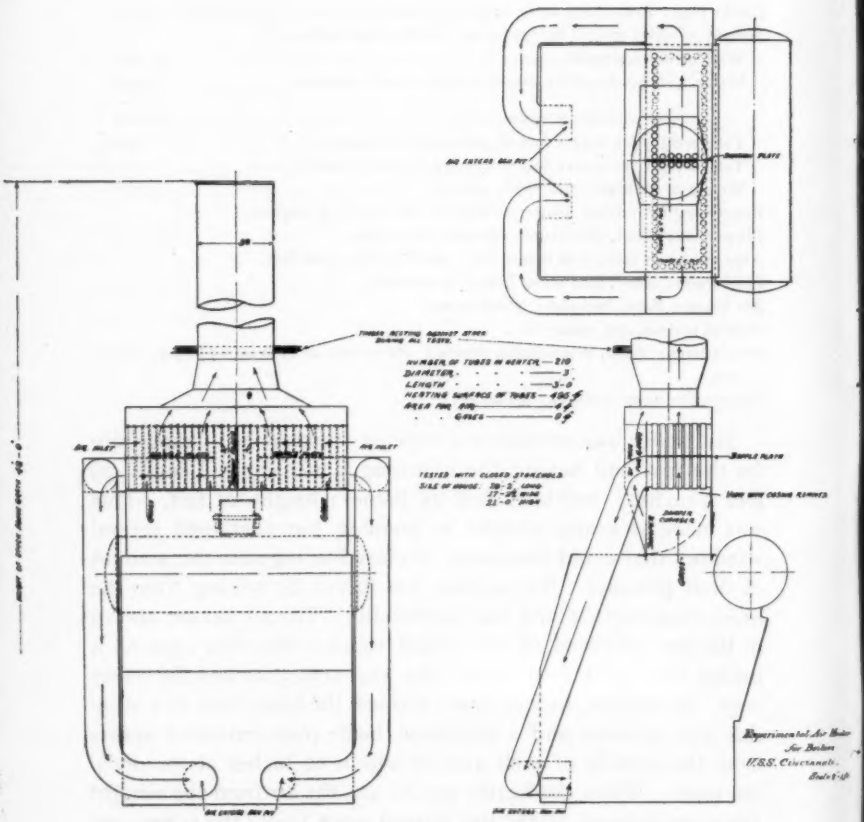
SECTION SHOWING PATH OF THE GASES.

Heating surface: Area, 2,640 square feet; ratio to grate, 1 : 41.74.
Per cent. water-heating surface, 100.
Grate bars: Kind, fixed.
Smoke pipe: Area, 7.876 feet; height, 48 feet above grate; ratio to grate, 1 : 8.03.
Weight of boiler and all fittings except uptakes and smoke pipe:

Without water, pounds	53,304
Water, 5 inches in glass; steam at 215 pounds, pounds	9,498
Total with water, pounds	62,802
Total weight per square foot of grate surface, pounds	992.9
Total weight per square foot of heating surface, pounds	23.79
Weight of air heater and ducts, pounds	5,320

Blower engines: Kind, blower driven by belt from shop engines.
Blower fan: Kind, Sturtevant; diameter, 60 inches.
Area of blower inlet, 9.62 square feet; outlet, 6.89 square feet.
Feed water: Kind, feed water heated by steam jet.
Air heater: Kind, two-pass; 3-inch tubes.
Area of surface, 495 square feet.
Feed pumps: Kind, Worthington duplex; dimensions of cylinders, 7½ by 4; 6-inch stroke.
Other boiler appurtenances: Steam jet.

The boiler was erected in a wooden structure built especially for the test and having the following dimensions: Length, 29 feet 2 inches; width, 17 feet 2½ inches; height, 21 feet. This was made as nearly airtight as possible, but contained several windows that could be opened or closed to regulate the amount of draft pressure. The blower was driven by belting from the main shop engines and ran continually. The air heater, shown in the cut, consisted of 210 3-inch tubes 3 feet long, spaced 4 inches from center to center one way and 5 inches the other way. A vertical division plate divided the tubes into two separate sets or nests, and a horizontal baffle plate extended across from the outside of each nest to within 12 inches of the division plate. When the heater was in use, the air from the airtight fire room entered at the top, passed once across the tubes over the baffle plate, and back under same to outlets leading to back of ash pit, as shown, the front ash-pit doors being closed. When the air heater was not to be used, the gases were bypassed by a damper, as shown on above drawing, and the air entered the ash pit through the front, the doors thereto being removed.


NUMBER OF TUBES IN HEATER — 210
DIAMETER — 3"
LENGTH — 3'-0"
HEATING SURFACE OF TUBES — 495 ☐'
AREA FOR GAS — 6 ☐'
" " GASES — 8 ☐'
TESTED WITH CLOSED STOKEHOLD.
SIZE OF GRATE: 78"-3" LONG
17'-2½" WIDE
21'-0" HIGH
BAFFLE PLATE
Experimental Air Heater for Boilers U.S.S. Cincinnati. Built 1898

ARRANGEMENT OF AIR HEATER.

DESCRIPTION AND OBJECT OF TESTS.

Seven tests were made in all. Six of these consisted of three pairs in which the two tests of each pair were under similar conditions in every way except that of using the air heater, one being with and the other being without this heater, in order to define the economy due to its use. The last or seventh test was for the maximum consumption, and was made without the air heater and with full grate. Two pairs of tests, one at a consumption of about 20 pounds of coal and the other at about 35 pounds of coal per square foot of grate per hour, were made with the full grate service in use. These tests will be found in tables of results numbered 1, 2H, 3H, 4, the letter H signifying that the air heater was in use during the tests. Tests 1 and 2H were of eight hours' duration, and tests 3H and 4 were of six hours' duration. The grate surface was then reduced to 52 square feet by a course and a half of bricks, seven courses in height, at back of furnace, and tests Nos. 5 and 6H, lasting fours each, were made, burning about 50 pounds of coal per square foot of grate per hour. The bricks were then removed from the furnace and test No. 7, lasting three and one-third hours, was made, burning nearly 60 pounds of coal per square foot of grate per hour. The data and results of these tests will be found in the accompanying tables.

COAL AND FIRING.

The coal used was Pocahontas coal from Flat Top Mine. It contained considerable slate and clinkered badly. On tests Nos. 1 and 2H run-of-mine coal was used; on tests Nos. 3H, 4, 5 and 6H the coal was screened, using a screen with a 1-inch mesh. On test No. 7 the screenings from the former tests were run over a $\frac{3}{8}$-inch mesh screen, and the coal thus screened was mixed with the screened coal used in other tests. The firing was good and very regular. Two alternate doors were fired in rapid succession. The other two sections of fires, in wake of the other two doors, were then leveled with a hoe, and then sliced through the slicing door, and then coaled, the average time between coalings of the same two furnaces being from eight to ten minutes. The furnace doors were open about twenty-five seconds when coaling and about ten seconds in leveling. The coal

made comparatively little smoke except when firing or working fires. The data in regard to smoke was taken by using Ringelmann charts.

DESCRIPTION OF APPARATUS.

The water was weighed in two tanks, each supported on a platform scales and run into a third tank below, from which the feed pumps drew water. There were two Worthington duplex, horizontal, outside-packed, plunger pumps. All pipes were above ground and in plain sight, and wherever connected to other piping or boilers, plugs were left out of T connections to show that there was no leakage. The gross and tare of each tank was taken, and the temperature was taken at the lower tank just as each upper tank drained into it. The feed water was heated by steam injection before entering the weighing tanks.

The coal was weighed in barrows on platform scales in the fire room and dumped on the floor. The time was taken when each lot of barrows were fired.

A sample shovelful of coal was taken from each lot of barrows and thrown into a barrel, and from this, mixed and quartered, the final samples for analyses, calorimeter and moisture determinations were taken. The gases for analyses were drawn from near the center of the base of smoke pipe by means of a pipe inserted therein connected with an aspirator and small Orsat instrument.

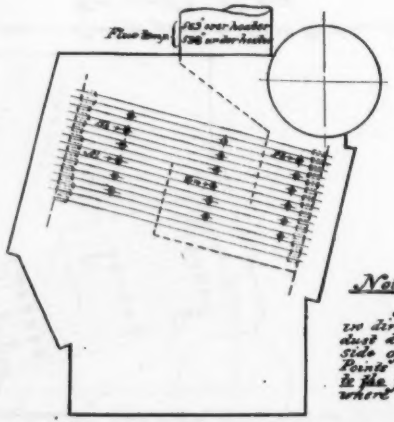
All draft pressures were taken outside the building, pipes being led there from the different places where pressure determinations were required.

Temperatures were taken at the back and front of uptake just above the heater; in front by a mercurial pyrometer, and at the back by a metallic pyrometer. When the air heater was used the temperature was taken in addition just below the heater by means of a mercurial pyrometer.

The moisture in the steam was determined by a Barrus universal calorimeter. The steam was found practically dry in all cases. The calorimeter was calibrated twice, and results will be found recorded in the tables. The steam was partly used in the shop and partly blown off into the atmosphere, the pressure being controlled by regulating a small stop valve by hand.

Melting Point of Metals

U.S.S. CINCINNATI.



Aluminum:- Al. 1160°

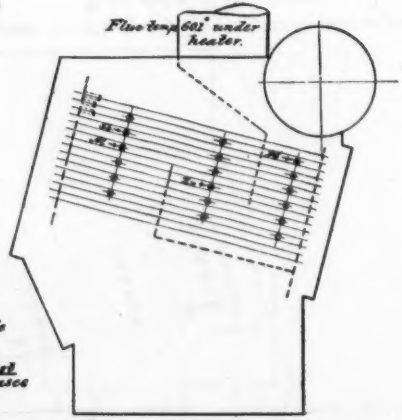
Antimony:- Sb. 842°

Zinc:- Zn. 780°

Lead:- Pb. 625°

Note!

Metals hung from rods placed in direct path of gases through side dust doors about four feet in from side of boiler.
Points marked show positions nearest to the furnace in the travel of the gases where metals melted.



June 18th 1900. Coal per sq. Grate 35.51 lbs.
Water per sq. H.S. (212°) 8.42.
With air heater.

June 19th 1900. Coal per sq. Grate 35.08 lbs.
Water per sq. H.S. (212°) 8.78.
Without air heater.

June 20th 1900. Coal per sq ft Grate 50.38 lbs (reduced grade)
Water per lb H.S. (212°) 10.07
Without air heater.

June 21st 1900. Coal per sq ft Grate 47.01 lbs (reduced grade)
Water per lb H.S.(212°) 9.58
With air heater.

Flue tube 3.68"
June 25th 1900
Coal per sq. ft. Grate 20.18 lbs.
Water per sq. ft. H.S. (212°) 5.57.
With air heater

Flue tube 3.68"
June 22nd 1900.
Coal per sq. ft. Grate 58.2 lbs. (full grate)
Water per sq. ft. H.S. (212°) 15.68.
Without air heater

Experiments to show the heat of the gases at various points were made by noting the points at which different metals melted. A small piece of metal was wired to a piece of ¼-inch pipe, and pushed in carefully through the sight doors at the side of casing to about the middle of the boiler; by noting where such metal would melt, and again introducing a piece of the same metal at another hole further along in the path of the gases until a position was reached when the metal would not melt, and by the use of various metals with known melting points, the temperature of the gases was determined and is plotted on the accompanying diagrams.

MEASUREMENT OF WATER IN BOILER, AND TIME OF GETTING UP STEAM.

Before making test No. 6H, on June 21, all water was drained from the boiler and the contents of boiler noted for each 1-inch mark of the water gage, with the following results:

TEMPERATURE OF WATER, 72 DEGREES FAHRENHEIT.

Height of water in gage.	Total water.	Difference.
Inches.	*Pounds.*	
0	9,312	. .
1	9,498	186
2	9,662	164
3	9,912	250
4	10,137	225
5	10,368	231
6	10,672	304
7	10,943	271
8	11,175	232

Fires were started in the boilers with lightwood and blower in use at 9.40 A. M. Temperature of water in boiler, 72 degrees; height in glass, 1 inch.

The following is a record of time and pressures or temperatures:

TEMPERATURE OF WATER, 72 DEGREES FAHRENHEIT.

Time.			Steam pressure.
By watch.	Elapsed.		
	Min.	*Sec.*	
9·40	0		Fires started.
9·45	5		Steam formed.
9·46·30	6	30	25 pounds.
9·47·30	7	30	35 pounds.
9·48	8		45 pounds.
9·48·30	8	30	55 pounds.
9·49	9		65 pounds.
9·49·30	9	30	75 pounds.
9·50	10		85 pounds.
9·50·30	10	30	95 pounds.
9·50·45	10	45	115 pounds.
9·51	11		125 pounds.
9·51·10	11	10	135 pounds.
9·51·15	11	15	145 pounds.
9·51·30	11	30	155 pounds.
9·51·40	11	40	165 pounds.
9·51·55	11	55	175 pounds.
9·52·10	12	10	185 pounds.
9·52·20	12	20	195 pounds.
9·52·30	12	30	205 pounds.
9·52·40	12	40	215 pounds.

An examination of the boiler after this test showed no injury or change in its condition in any respect.

DATA OF TESTS.

Number of tests.	1.	2H.	3H.	4.	5.	6H.	7.
Date of tests, 1900	June 15.	June 16.	June 18.	June 19.	June 20.	June 21.	June 22.
Duration of test, hours	8	8	6	6	4	4	3½
Kind of start	Alternate.	Alternate.	Alternate.	Alternate.	Alternate.	Alternate.	Alternate.
State of weather	Clear.	Clear.	Clear.	Clear.	Clear.	Clear.	Clear.
AVERAGE PRESSURES.							
Barometer, inches	.....	30 18	30.12	30.22	30.20	30.09	30.02
Steam pressure by gage, pounds	209.3	206.6	210 6	207.4	213.6	209.8	210.8
Force of draft at base of pipe, inches of water	.097	.36	.079	.127	.34	.37	1.48
Force of draft in furnace, inches of water	.098	.056	.81	.967	.99	.65	.54
Force of draft in ash pit, inches of water	.25	.107	1.42	1.37	1.75	1.45	1.685
AVERAGE TEMPERATURES.							
External air, degrees Fahrenheit	.....	84.7	80.2	75.8	80	85	90.7
Fire room, degrees Fahrenheit	101.5	99.5	98.2	101.8	109.3	115.4	117.0
Steam	391.4	390.4	391.9	390.6	393.0	391.6	391.9
Air entering heater, degrees Fahrenheit	.....	93.4	94.1	.....	.....	106.1	.....
Air leaving heater, degrees Fahrenheit	.....	206.2	235.9	.....	.....	259.5	.....
Feed water entering boiler, degrees Fahrenheit	107.5	106.2	103	101.8	103.3	102.0	99.7
Air entering ash pit, degrees Fahrenheit	101.5	206.2	235.9	101.8	109.3	259.5	117
Escaping gases from boiler, degrees Fahrenheit	466	493	608.6	570.6	640.0	637	Less than 900° F.
Escaping gases from heater, degrees Fahrenheit	.....	458	569	.....	.....	674	.....
FUEL.							
Kind, Pocahontas coal	Run of mine.	Run of mine.	Screened.	Screened.	Screened.	Screened.	Twice screened.
Weight of coal as fired, pounds*	10,345	10,600	12,925	13,517	10,600	10,046	12,600
Moisture in coal, per cent	4.06	3.62	1.6	1.5	1.13	1.0	0.88
Weight of dry coal consumed, pounds	9,925	10,201	12,718	13,314	10,480	9,946	12,489
Weight of ash and refuse, pounds	996	879	1,088	1,268	1,177	949	1,055
Weight of combustible consumed, pounds	8,929	9,322	11,630	12,046	9,303	9,097	11,434
Refuse in dry coal, per cent	10.03	8.60	8.55	9.52	11.23	9.54	8 45
FUEL PER HOUR.							
Coal consumed per hour, pounds	1,293.1	1,325	2,154.2	2,252.8	2,650	2,512.5	3,780
Dry coal consumed per hour, pounds	1,240.6	1,277	2,119.7	2,219	2,620	2,486.5	3,747
Combustible consumed per hour, pounds	1,116 1	1,157	1,938.3	2,007.7	2,325.7	2,247.3	3,430
Coal consumed per hour per square foot G.S., pounds	20.45	20 95	34.06	35.61	50 95	48.31	59.75
Dry coal consumed per hour per square foot G.S., pounds	19.61	20.19	33 51	35.08	50 38	47 81	59.23
Combustible consumed per hour per square foot G S., pounds	17.65	18 29	30.64	31.74	44.72	43.73	54.22
Coal per hour per square foot H.S., pounds	.490	.502	.816	.853	1.004	.592	1.432
Dry coal per hour per square foot H.S., pounds	.470	.484	.803	.840	.992	.942	1.419
Combustible per hour per square foot H.S., pounds	.423	.438	.734	.760	.881	.861	1.299

* Including equivalent of wood used in lighting fires.

DATA OF TESTS.—CONTINUED.

Number of tests.	1.	2H.	3H.	4.	5.	6H.	7.
QUALITY OF STEAM.							
Per cent. of moisture in steam	0	0	0	0	.25	.16	0
Degrees of superheating	0	0	0	0	.....	.....	.....
Quality of steam (dry steam = 100)	100	100	100	100	99.75	99.84	100
WATER.							
Total weight of water fed to boiler, pounds*	93,928	100,783	113,943	118,330	91,066	86,488	102,545
Water actually evaporated, corrected for quality of steam, pounds	93,928	100,783	113,943	118,330	90,838	86,350	102,545
Factor of evaporation	1.1657	1.1667	1.1705	1.171	1.1704	1.1715	1.173
Equivalent water evaporated into dry steam from and at 212°, pounds	109,492	117,583	133,370	138,586	106,317	101,159	120,285
WATER PER HOUR.							
Water evaporated per hour, corrected for quality of steam, pounds	11,741	12,598	18,974	19,722	22,709	21,622	30,763.5
Equivalent evaporation from and at 212°	13,686	14,698	22,228	23,098	26,579	25,290	36,086
Equivalent evaporation from and at 212° per square foot G.S., pounds	216.3	232.4	351.4	365.2	511.1	486.3	570.5
Same per square foot of heating surface, pounds	5.18	5.57	8.42	8.75	10.07	9.58	13.67
ECONOMIC RESULTS.							
Water apparently evaporated under actual conditions per pound of coal as fired, pounds	9.079	9.508	8.816	8.754	8.591	8.609	8.138
Apparent equivalent evaporation from and at 212° per pound of coal (including moisture), pounds	10.58	11 09	10.32	10.25	10.03	10.07	9.545
Equivalent evaporation from and at 212° per pound of dry coal, pounds	11.03	11.51	10.49	10.41	10.14	10.17	9.63
Equivalent evaporation from and at 212° per pound of combustible, pounds	12.19	12.70	11.47	11.50	11.43	11.12	10.52

* Corrected for inequality of water level and steam pressure at beginning and end of test.

In tests marked H the draft heater was in use.

REMARKS AND OBSERVATIONS.

Principal data taken every fifteen and thirty minutes.

Percentage of smoke as observed, using Ringelmann smoke charts: Test 1, No. 0 to No. 1 and as high as No. 4 when firing; test 2H, No. 0 to No. 1, with puffs of No. 4 when firing; test 3H, No. 1 to No. 2, with puffs of No. 4 when firing; test 4, No. 2 to No. 3, with puffs of No. 4 when firing; test 5, No. 1 to No. 3, with puffs of No. 4 when firing; test 6, about No. 2, with puffs of No. 4 when firing; test 7, average about No. 3, with puffs to No. 5 when firing.

Kind of firing: Spreading and slicing.

Average thickness of fires: Tests 1 and 2H, 6 inches; test 3H, 7 inches; tests 4, 5, 6 and 7, 6 inches.

Average intervals between firings for each furnace during time fires were in normal condition: Eight to ten minutes.

Average interval between times of breaking up: Eight to ten minutes.

Efficiency of firemen: Expert.

ANALYSES OF WASTE GASES MADE DURING TESTS OF U. S. S. *CINCINNATI* BOILER, ELIZABETHPORT, N. J., JUNE, 1900.

Date.	Time.	Condition of fire when sample was taken.	CO^2.	O.	CO.	Pounds dry gas per pound carbon.
1900.	4'58		15.2	3.3	1.0	
	5'15		14.3	3 0	2.0	
June 15.	5'30	Just before firing	13.0	6.5	0.0	
	5'55	One minute after firing	12 5	6.7	0.8	
	6'16	Just after raking	14.3	3.7	1.0	16.8
	6'27	Two minutes after firing	12.7	6 6	0 7	
	7'05	Three minutes after raking and just before firing	16.0	2.0	2.0	
		Average	14.0	4.5	1.1	
	11'45		13.4	6.4	0.0	
June 16.	12'50	Just after firing	12.0	5.0	1.0	
	1'50	Just after slicing	12.0	6.6	0.2	19.1
	3'50	Just after slicing	13.2	4.8	0.7	
		Average	12.7	5.7	0.5	
	11'25	One-half minute after firing	12.3	3.4	2.7	
	12'40	While slicing	14.2	4.0	0.1	
June 18.	12'50	Just after slicing	12.5	4.3	1.2	
	12'58	Just before slicing	13.0	4.0	3.4	17.3
	1'03	One minute after firing	13.5	5 4	0.2	
		Average	13.1	4.2	1.5	
	10'10	While slicing	15.0	3.2	1.2	
	10'25	While slicing (all samples except 11 o'clock collected through ¼-inch iron pipe)	13 8	5.2	0.6	
	10'28	Just after raking	14 4	3.1	0.9	
June 19.	10'35	One minute before firing	13.2	5.6	0.4	
	11'00	While slicing (sample collected through glass tube)	13.0	5.6	0.6	18.8
	2'20	Just after raking	10.2	8.3	0.5	
	2'40	Just after firing	10 2	9.0	0.3	
		Average	12 8	5.7	0 6	
	10 25	While slicing	13 5	5.7	0.0	
	11'00	While slicing	11.2	8.4	0 3	
June 20.	11'04	Just after firing	10.4	8.1	0.5	
	11'13	Two minutes before raking	9.2	9.9	0 0	20.6
	12'25	Just after raking	12.1	5.4	0 7	
	12'36	Just after raking	14.2	4.0	0 8	
		Average	11.8	6.9	0 4	
	11'00	Just after raking	15 7	4.6	0.1	
	11'03	One minute before raking	13.0	6 0	0.0	
June 21.	11'13	Just after firing	15.4	3.0	0.6	
	11'50	Just after raking	13.6	5.6	0 1	17.7
	11'55	One minute before raking	13.0	5.3	0.4	
	11'59	Just before raking	16.0	4.2	0.0	
		Average	14.5	4.8	0.2	
	11'21	Two minutes before firing	14.3	4.2	1.1	
	11'45	Two minutes before firing	11 0	9.0	0.0	
June 23.	12'38	Just before leveling and firing	...	...	...	
	2'26	Just after firing	11.8	7 9	0.4	18.6
	2.30	Just after firing	13 3	4.2	1.0	
	2'43	Two minutes before firing	14.2	3 8	1.0	
		Average	12.9	5 8	0.7	
	10'16	Just before firing	15.3	4.1	1.0	
	10'21	One minute before firing	13 0	6.0	1.0	
June 25.	11'10	Just after leveling	13 7	6.6	0.3	
	11'13	Just after firing	14.0	5.2	0.8	18.5
	11'47	Just after firing	9.0	11.2	0.3	
		Average	13.0	6.6	0.7	

COAL-CONSUMPTION TESTS OF S. S. *JOHN W. GATES.*

By Lieutenant B. C. Bryan, U. S. Navy.

Between October 10 and 15, 1900, tests were made on the Lake Steamer *John W. Gates*, owned by the American Transportation Co., by Lieutenant Commander J. H. Perry and Lieutenant B. C. Bryan, U. S. Navy.

The object of these tests was primarily to note the action of and results obtained using the Crowe Patent Stoker.

Four tests in all were made, of ten, four, eight and six hours' duration, respectively. During the tests indicator cards were taken from the main engines, and the usual observations of pressures and temperatures recorded. The coal was carefully weighed and logged on each test.

Tests Nos. 1 and 2 were made with the vessel light, on the up trip, in Lakes Huron and Superior, respectively. Test No. 1 was made under the usual running speed of the vessel when light, and amounted to merely weighing coal and taking observations for ten hours out of the run. Test No. 2 was made using a steam jet in the smoke pipe to increase the draft.

Tests Nos. 3 and 4 were made on the down trip, after having loaded at Two Harbors, Minn., with about 7,000 tons of ore, the vessel drawing about 17 feet 10 inches of water. Test No. 3 was made at the usual running speed, and Test No. 4 with draft increased by steam jet in smoke pipe.

The principal data of the vessel and machinery follow, but a few observations here are not considered out of place. This machinery was built under the supervision of the able Chief Engineer of the American Transportation Co., Mr. Joseph F. Hayes, and the great economy obtained is largely due to his care in the design and arrangement of the plant. The ratio of high to low-pressure cylinder area is 1 to 13.22. Joy valve gear is used on the high and intermediate-pressure cylinders, giving in the high-pressure cylinder an admission of steam almost per-

fect, as is shown by the indicator cards therefrom. The cylinder ports are made large, while the clearance is reduced as much as possible. A feed heater is provided, into which all the auxiliaries necessary for heating the feed water are exhausted. The dynamo when running exhausts into the third receiver of the main engine, and all precautions have been taken to make these engines economical, and with great success, as is shown by the results.

The type of Babcock and Wilcox boiler adopted is one that recent tests made by government officials, given in this number (see page 882), show to be exceedingly economical under various conditions.

During the tests on the *Gates* the only auxiliaries independent of the main engines in use were the air pumps, feed pumps, stoker engines, steering engines, and at occasional times the dynamo engines. The stoker engines exhausted into the atmosphere, and they used such a small amount of steam that, under the circumstances, this seems to be warranted to avoid a complication of piping.

It is much to be regretted that the actual consumption of water per indicated horsepower was not measured on these trials, time not allowing arrangements to be made for this purpose, but very accurate approximation can probably be made by comparing this with the trial of the U. S. S. *Pennsylvania*, given in Vol. II, No. 3, of the JOURNAL, and from the results of evaporation tests of this type of boilers as given in this number. The latter tests show that with the coal used thereon 9.079 pounds of water were evaporated per pound of coal as fired under actual conditions when burning 20.45 pounds of coal per square foot of grate, the factor of evaporation being 1.165, and the coal used containing 4.06 per cent. of moisture, or about the same amount of moisture as the coal used on the tests of the *Gates*.

Taking the test made on the *Gates* on October 10, 1900, burning 20.62 pounds of coal per square foot of grate, the temperature of feed entering boiler was 186 degrees, the average steam pressure was 244. Factor of evaporation was 1.0876, while in the evaporation tests of same type of boiler referred to above the factor of evaporation was 1.165 per cent. The proportion of heating surface to grate was, in the case of *Gates*,

equal to 55.55, while in the boiler on which the evaporative tests were made the proportion was 41.74. It is probable therefore, that a greater proportion of the total heat was utilized in the trial of the *Gates* than in the evaporation tests referred to. A calorimeter test of the coal used on the trial of the *Gates* gave 13,580 B.T.U. (See end of this article.) The coal used on the evaporative tests was of a better quality and probably contained at least 14,000 B.T.U. Supposing the efficiency of the boiler the same on each test, the expected evaporation per pound of coal as fired would be $9{,}079 \times \frac{1.165}{1.0876} \times \frac{13580}{14000} = 9.43$ pounds.

The total amount of coal burned per hour was 2,227 pounds, which would give $2{,}227 \times 9.43 = 21{,}000$ pounds of water evaporated per hour.

The air pump was the same type and make as that of the U. S. S. *Pennsylvania* when the steam per I.H.P. per hour was 83.18. The cards taken from the air pump of the *Gates* showed 6.66 I.H.P. Taking the same consumption of water per I.H.P., the hourly consumption would have been $83.18 \times 6.66 = 554$ pounds.

The feed pump was of the same type, make and size as that of the *Pennsylvania*, which used, at 1,455 I.H.P. of main engines, 595 pounds of water per hour. The dynamos on the *Gates* were compound, and probably more economical than those of the *Pennsylvania*, the port dynamo of which latter vessel used 480 pounds of steam per hour.

The steering engine of the *Gates* was slightly larger than that of the *Pennsylvania*, but the vessel steered better and probably used about the same amount of steam, viz: 125 pounds per hour.

We have then for steam used by auxiliaries, average per hour:

	Pounds.
Air pumps,	554
Feed pumps,	595
Stoker engines (from actual test made),	140
Dynamos in operation 3 hours $= 480 \times 3 \div 10 =$	144
Steering engines,	125
Total,	1,558

Subtracting this from the 21,000 pounds evaporated by boilers, we have 21,000 — 1,558, or 19,444 pounds of steam used by the main engines, or $\frac{19,444}{1430.7} = 13.59$ pounds of water per I.H.P. per hour for the consumption of the main engines.

The principal dimensions of the hull and machinery are as follows:

HULL.

The hull of this vessel was built at the Cleveland Shipbuilding Co.'s works at Lorain, Ohio. The framing and deck beams are made from channel steel and the vessel is well stiffened by longitudinal girders and stringers. Transverse bulkheads divide the vessel into thirteen water-tight compartments. There are fifteen cargo hatches 8 feet by 30 feet, spaced 24 feet between centers.

Length over all, feet,	498
Beam, feet,	52
Depth, feet,	30
Height of water bottom, feet,	6
Capacity of water bottom, tons,	3,500
Bunker capacity, about, tons,	300

ENGINES.

The engines are placed at the extreme end of the vessel. The boilers are immediately forward of the engines, with an athwartship coal bunker forward of the boiler-room bulkhead.

The main propelling engine is of the vertical, direct-acting, inverted, jet-condensing, quadruple-expansion type. One piston valve actuated by a Joy valve gear, is fitted on the side of each of the high, first and second intermediate pressure cylinders. The low-pressure valve is a double-ported slide, placed on the after side of the cylinder, and operated by eccentrics and a double-bar Stephenson link.

Number of cylinders, H.P., inches	16½
1st I.P., inches	25
2d I. P., inches	38½
L.P., inches	60
Stroke, inches	40
Diameter of piston rods (one for each cylinder), inches	4¾
Order of cylinders from forward	(1) H.P; (2) 1st I.P.; (3) 2d I.P.; (4) L.P.
Sequence of cranks	H.P.; L.P.; 1st; 2d.

The H.P. and 1st I.P. are at 180 degrees, as are the 2d I.P. and L.P., the former being at 90 degrees with the latter.

There is one four bladed propeller, 14 feet in diameter with 15 feet 6 inches pitch.

BOILERS.

Steam is supplied by two Babcock and Wilcox boilers of their latest Marine type, having baffle plates directing the products of combustion three times across the tubes before leaving the boilers. (See page 882 of this number for illustration of this type of boiler.) Each boiler is 10 feet long, 10½ inches wide, and 13 feet 10 inches high, containing 3,000 square feet of heating surface and suitable for from 65 to 70 square feet of ordinary grate surface for hand firing. The total grate surface of all stokers is 108 square feet.

The weight of the two boilers dry is 109,260 pounds, and with water, 132,590 pounds.

The bottom and top rows of tubes are 4 inches in diameter and all others are 2 inches in diameter. All tubes are of seamless cold-drawn steel, the 4-inch tubes being No. 6, B.W.G., and the 2-inch tubes, No. 10, B.W.G. in thickness. The lengths between headers is 9 feet.

AUXILIARY MACHINERY.

INDEPENDENT AUXILIARY MACHINERY.

Number.	Purpose.	Make.	Type.	Diameter of cylinders. H.P.	L.P.	Water.	Stroke.	Reference number.
				In.	*In.*	*In.*	*In.*	
1	Main air pump	Blake	Vertical, compound, simplex	11	18	22	18	A
1	Main feed pump	do	Hor., comp., duplex, plunger	6	10	5	10	B
1	Aux. feed and ash pump	do	Hor., duplex, plunger	...	8	4	10	C
2	Ballast pumps	do	Hor., comp., duplex	7½	12	16	12	D
1	Deck pump	do	Hor., duplex	...	4½	3½	6	E
2	Dynamo engines	Buffalo Forge Co.	Tandem, compound	4½	7	...	6	...
1	Steering engine	Clev'd S. B. Co.	Hor., two-cylinder	...	8	...	8	...
1	Windlass engine	do	do	...	...	...	...	...
1	Capstan engine	do	do	...	...	...	...	...
5	Deck winches	do	do	...	8	...	10	...
1	Ventilating blower	Sturtevant	Vertical, single-cylinder	...	3	..	4½	...
4	Stoker engines	.....	Single-cylinder, oscillating	...	...	...	...	...

Other Auxiliaries.

There is one bilge pump, 6 inches by 10 inches, and one cooler pump, 3 inches by 12 inches, worked from the low-pressure cross-head, both being single-acting plunger pumps. There is also a feed heater which receives the exhaust of all auxiliary machinery except dynamo engine. There are also two purifiers over boilers.

STOKERS.

Two mechanical stokers of the Crowe pattern were fitted to each boiler. This stoker consists, essentially, of a set of bars carried from front to back of the furnace, over a number of fair leaders, by two chains, one on each side of the furnace. At the back of the furnace the chains and bars pass over a drum and thence back over fair leaders to the front of the furnace again. At the front of the furnace the chains pass over sprockets from which they receive motion. The bars each have two wrought-iron lugs on lower edge which hook into the links of the chain, alternate bars hooking in from opposite sides. The upper faces of the bars are toothed and locked into each other, leaving sufficient opening to form air spaces for burning the coal. From this construction it is evident that a bar can only be removed and replaced when it is passing over the sprocket wheel in the front of the furnace, for at no other time are the teeth entirely disengaged from each other.

The sprockets at the fronts of the furnaces, which give motion to the chains and bars, may be driven by any suitable device, but as the motion is so very slow, about 6 feet an hour when burning 20 pounds of fuel per square foot of grate, worm gearing is best employed. At the front of the furnace a hopper extends the whole width, into which coal is shoveled.

On the boiler front is a distributing and regulating plate which extends across the entire width of the grate and is raised up above it to whatever thickness it is desired to carry the fire. At the front of the furnace is a brick arch, and at the back there is an apron plate, through which are a number of holes, to allow a supply of air to enter over the fire.

The speed of the bars from front to back is so regulated that coal shall be entirely consumed in the passage, and this speed will of course vary with the draft and the thickness of fire carried. The fires are worked entirely from the back end, the front being tightly closed in, with the object of excluding all air from over the fire. The ashes, clinker and unconsumed coal are dumped in the ash pan at the back end of the furnace.

While under way the fire required nearly constant attention, as small clinkers would form, which would be caught by the plate at the back of the furnace and combed back over the fire. These clinkers would constantly increase, until finally the whole air space in the back was choked by them and they had to be removed with a pricker bar. If the speed of the grate is too low, the fire burns out before reaching the back end of the furnace, allowing cold air to enter, which kills the effect of the fire.

Clinkers, forming on the brick sides of the furnace, interfere to some extent with the working of the stoker.

To build up the fire, or to pass from one speed to a higher one with the engine, requires from one-half to three-quarters of an hour, depending on the air pressure and speed of the bars.

During the entire trip the stokers worked satisfactorily. During most of the time little or no smoke was emitted from the pipe except while the fires were being worked from the back, or when an additional amount of coal worked in under the plate in the front of the furnace. The air pump worked regularly and quietly, but for some reason, probably due to the large clearance required in the cylinders of this type of pump, the vacuum carried was not much in excess of 23½ inches.

Lead did not melt during any of the tests when suspended in the uptakes just over the top row of 4-inch tubes or practically where the gases leave the boiler proper.

Lead suspended in the boiler where the gases leave the last row of 2-inch tubes melted on the test of October 15, but only softened on the tests of October 10 and 13.

The following tables, Nos. I, II, III and IV, give the data of the tests:

TABLE I.—OCTOBER 10, 1900, 9·30 A. M. TO 7·30 P. M.

Duration of Test 10 hours.

Time	Counter	Revs	Steam: Boiler	Steam: 1st Rec.	Steam: 2nd Rec.	Steam: 3rd Rec.	Vacuum	Throttle	Temperature: E.R.	Temperature: Injector	Temperature: Disc'g.	Temperature: Feed	Double Strokes: Air Pump H.P.	Double Strokes: Air Pump L.P.	Double Strokes: Feed Pump	Air Pressure	Coal	Indicated H.P. from full indicator diagrams: H.P	1st I.P	2nd I.P	L.P
9 00	377785		245	110	33	7	23½	10	80	63	114	176	22	19		25		3	3.5	3.5	4.5
10 00	380260	82.5	245	108	33	7	24	10	82	63	114	175	22	19		25		3	3.5	3.5	4.5
10 30	382740	82.7	245	108	33	7.5	24	10	80	63	114	185	22	19	18	25		3	3.5	3.5	4.5
11 00	385225	82.8	245	110	33	7.5	24	10	80	62	113	188	22	19		25		3	3.5	3.5	4.5
11 30	387710	82.8	245	108	33	7.5	24	10	82	61	112	190	22	19	18	.3		3	3.5	3.5	4.5
12 00	390195	82.8	245	110	33	7.5	24	10	82	62	114	196	22	19		.3		3	3.5	3.5	4.5
12 30	392685	83	245	110	33	7.0	24	10	82	62	114	190	22	19	19	.3		3	3.5	3.5	4.5
1 00	395150	82.2	240	105	32	7	24	10	82	62	114	190	22	19		.3		3	3.5	3.5	4.5
1 30	397640	83	245	108	33	7.5	24	10	82	62	114	190	22	19	19	.3		3	3.5	3.5	4.5
2 00	400115	82.5	240	105	31	7	24	10	82	62	110	184	22	19		.3		3	3.5	3.5	4.5
2 30	402565	81.7	236	106	31	6.5	24	10	86	62	110	180	22	19	18	.3		3	3.5	3.5	4.5
3 00	405075	83.7	245	108	32	7	24	10	88	62	115	185	22	19		.3		3	3.5	3.5	4.5
3 30	407550	82.5	245	107	32	7	24	10	88	62	114	186	22	19	19	.3		3	3.5	3.5	4.5
4 00	410040	83	245	108	33	7.5	24	10	88	62	114	182	22	19		.3		3	3.5	3.5	4.5
4 30	412515	82.5	245	108	33	8	24	10	88	62	114	180	22	19	18	.3		3	3.5	3.5	4.5
5 00	415010	83.1	245	108	33	8	24	10	88	62	114	180	22	19		.3		3	3.5	3.5	4.5
5 30	417480	82.3	245	105	31	7	24	10	85	62	113	184	22	19	18	.3		3	3.5	3.5	4.5
6 00	419970	83	245	110	32	8	24	10	85	62	118	186	22	19		.3		3	3.5	3.5	4.5
6 30	422465	83.2	245	108	32	8	24	10	80	61	114	190	22	19	18	.3		3	3.5	3.5	4.5
7 00	424950	82.5	245	110	33	8	24	10	80	60	112	190	22	19		.3		3	3.5	3.5	4.5
7 30	427445	83.2	245	108	32	8	24	10	80	60	114	188	22	19	18	.3		3	3.5	3.5	4.5
		82.77	244	107.8	32.4	7.5	24	10	83.5	61.3	113.5	185.5	22	19	18	.8		3	3.5	3.5	4.5

Total = 22270 lbs. Per hour = 2227 Per sq. ft. grate per hour = 2062 Ashes = 3750 lbs = 17% Moisture = 04%

I.H.P during run	1,430.7
Pounds coal per I.H.P. per hour, as fired	1.56
Pounds coal per I.H.P. per hour, dry	1.5

Auxiliaries in operation during whole test, air pump, feed pump, stoker engines.
Also dynamo engine from 4·30 to 7·30, exhausting into third receiver.

TABLE II.—OCTOBER 11, 1900. 2·15 TO 6·15 P. M.

Duration of Test 4 hours.

Time	Counter	Revs.	Steam: Boiler	Steam: 1st Rec.	Steam: 2nd Rec.	Steam: 3rd Rec.	Vacuum.	Throttle	Temperature: F.R.	Temperature: Injector	Temperature: Disch'g	Temperature: Feed.	Double Stroke: Air Pump H.P.	Double Stroke: Air Pump L.P.	Double Stroke: Feed Pump	Air Pressure	Coal	Link in Full throw: H.P.	Link in Full throw: 1st I.P.	Link in Full throw: 2nd I.P.	Link in Full throw: L.P.
2.10"	504480		250	115	34	8	24	10	85	55	114	175	25	22	22	.6	P. M. 363 Tn	.75	1.5	1.75	1.75
2.40"	507205	90.8	250	115	34	8	24	10	85	55	115	175	25	22	22	.6	P. M. 3364	.75	1.5	1.75	1.75
3.10"	509910	90.2	250	115	34	8	23	10	83	55	115	180	25	22	22	.6	Ashes 1%	.75	1.5	1.75	1.75
3.40"	512610	90	240	108	33	7.5	23	10	83	54	112	180	23	22	25	.5	Coal 4535 lbs	.75	1.75	1.75	1.75
4.10"	515280	89	235	108	33	7.5	23	10	83	54	112	180	23	22	25	.5	P. per sq ft grate per hr. 33.64	.75	2.25	1.75	1.75
4.40"	517980	90	245	117	33	8	23	10	81	53	115	180	23	22	25	.6	Moisture .54% Ashes 1%	.75	2.25	2.25	1.75
5.10"	520680	90	245	116	34	8	24	10	80	52	110	185	23	22	25	.6		.75	2.25	2.25	1.75
5.40"	523350	89.2	240	115	35	8	23	10	84	52	118	182	23	22	25	.6		.75	2.25	2.25	1.75
6.10"	526030	89.2	240	116	35	8	23	10	80	52	114	180	23	22	25	.6		.75	2.25	2.25	1.75
Ave.	21550		243.9	113.9	34.1	7.9	23.3	10	82.7	53.6	113.9	179.7	23.7	22	24	.58					
		89.8		122.6	48.5	21.6															

I.H.P. during test..... 1,818.3
Pounds coal per I.H. per hour, as fired..... 1.998
Pounds coal per I.P. per hour, dry..... 1.92

Auxiliaries in operation during whole test, air pump, feed pump, stoker engine.
Also dynamo engine from 4·15 to 6·15 P. M., exhausting into third receiver. Jet in smoke pipe.

TABLE III.—OCTOBER 13, 1900, 9·15 A. M. TO 5·15 P. M.

Duration of Test- 8 hours.

Time	Counter	Revs.	Steam: Boiler	Steam: 1st Rec.	Steam: 2nd "	Steam: 3rd "	Vacuum	Throttle	Temperature: F.R.	Temperature: Injection	Temperature: Disc'g.	Temperature: Feed	Double Strokes: Air Pump: H.P.	Double Strokes: Air Pump: L.P.	Double Strokes: Feed Pump	Air Pressure	Coal	Links in F.U. from Throw: H.P.	Links in F.U. from Throw: 1st I.P.	Links in F.U. from Throw: 2nd I.P.	Links in F.U. from Throw: L.P.
9.15	671750		250	106	33	2	24	10	80	50	110	192	19	16	21	.3		3.5	3.75	3.75	3.75
9.45	674090	78	250	105	32	6	24	10	80	50	144	86	19	16	21	.3		3.5	3.75	3.75	3.75
10.15	676415	77.8	250	105	32	5	23.5	10	80	50	116	188	19	16	20	.3		3.5	3.75	3.75	3.75
10.45	678755	78	250	105	32	6	23.5	10	80	50	116	186	19	17	20	.3		3.5	3.75	3.75	3.75
11.15	681085	77.7	250	105	32	6	23.5	10	80	50	116	190	19	18	19	.35		3.5	3.75	3.75	3.75
11.45	683420	77.8	250	105	32	6	23.5	10	80	50	115	185	19	18	18	.35		3.5	3.75	3.75	3.75
12.15	685750	77.7	250	105	32	6	23.5	10	80	50	114	184	20	18	18	.35		3.5	3.75	3.5	3.75
12.45	688085	77.8	250	105	32	6	23.5	10	80	50	114	182	20	18	18	.3		3.5	3.75	3.75	3.75
1.15	690405	77.5	245	104	31	5	23.5	10	80	50	114	180	20	18	18	.3		3.5	3.75	3.75	3.75
1.45	692700	76.6	240	105	31	6	23.5	10	80	50	115	186	20	18	18	.3		3.5	3.75	3.75	3.75
2.15	695030	77.7	245	110	34	7	23	10	80	50	120	190	22	20	18	.3		3.25	3.75	3.75	3.75
2.45	697370	78	245	110	34	7	23	10	80	50	122	192	23	21	18	.3		3.25	3.75	3.75	3.75
3.15	699715	78.2	245	110	34	7	23	10	80	50	125	194	23	21	18	.35		3.25	3.75	3.75	3.75
3.45	702076	78.7	250	115	35	8	23	10	80	50	125	190	23	21	18	.35		3.25	3.75	3.75	3.75
4.15	704390	77.2	245	110	34	7.5	23	10	80	50	120	188	22	21	18	.35		3.25	3.75	3.75	3.75
4.45	706750	78.7	250	112	35	8	23	10	80	50	118	190	22	20	18	.35		3.25	3.75	3.75	3.75
5.15	709115	78.8	250	112	35	8	23	10	80	50	112	180	22	20	18	.35		3.25	3.75	3.75	3.75
	37365																				
	77.84		245.8	107.7	32.9	6.5	23.3	10-	80	50	117.3	187	20.1	18.8	18.7	.325					

Coal = 17094. Coal per hour, 2137.7 lbs.
Ash = 1192. Moisture = 949 lbs.
Sp. gr. of coal = 1.27

I.H.P. during run..... 1,343.6
Pounds coal per I.H.P. per hour, as fired..... 1.59
Pounds coal per I.H.P. per hour, dry..... 1.53

Auxiliaries in operation, air pump, feed pump and stoker engines during whole time.
Also dynamo engine from 4·20 to 5·15, exhausting into third receiver.

TABLE IV.—OCTOBER 15, 1900. 7 A. M. TO 1 P. M.

Duration of Test 6 hours.

Time.	Counter.	Revs.	Steam. Boiler.	Steam. 1st Rec.	Steam. 2nd Rec.	Steam. 3rd Rec.	Vacuum	Throttle.	Temperature. F.R.	Temperature. Injector.	Temperature. Dischg	Temperature. Feed	Double Strokes. Air Pump. H.P.	Double Strokes. Air Pump. L.P.	Double Strokes. Feed Pump	Air Pressure.	Coal	Lifts of Fulcrum. H.P.	Lifts of Fulcrum. 1st I.P.	Lifts of Fulcrum. 2nd I.P.	Lifts of Fulcrum. L.P.
7.00	844330																Coal = 20655 lbs. Sq. ft grate = 31.77 Pr. hr = 3442.5 Moisture = 04 % Ashes = 17%	1	1	1.5	2.25
7.30	846890	85.3	250	108	34	9	23	10	74	61	115	174	19	17	23	.6		1	1	1.5	2.25
8.00	849455	85.5	250	108	34	9	23	10	74	61	115	180	19	17	23	.6		1	1	1.5	2.25
8.30	852000	84.8	250	108	34	9	23	10	76	61	115	186	19	16	24	.6		1	1	1.5	2.25
9.00	854560	85.3	250	109	34	9	23	10	76	60	115	186	19	16	24	6		.75	1	1.5	2.25
9.30	857130	85.7	250	109	34	9	23	10	76	60	115	190	19	16	24	.6		.75	1	1.5	2.25
10.00	859710	86.0	250	109	34	9	23	10	76	61	115	186	19	16	24	.6		.75	1	1.5	2.25
10.30	862265	85.2	250	108	34	9	23	10	76	62	116	188	19	16	24	.6		.75	1	1.5	2.25
11.00	864830	85.5	250	109	34	9	23	10	76	62	116	189	19	16	24	.6		.75	1	1.5	2.25
11.30	867385	85.2	250	109	34	9	23	10	76	62	116	188	19	16	24	.6		.75	1	1.5	2.25
12.00	869940	85.2	250	109	34	9	23	10	78	62	116	193	19	16	24	.55		.75	1	1.5	2.25
12.30	872500	85.3	250	109	34	9	23	10	78	62	115	188	19	16	24	.6		.75	1	1.5	2.25
1.00	875060	85.3	250	109	34	9	23	10	78	62	115	190	19	16	24	.6		.75	1	1.5	2.25
	30730																				
	85.36		250	108.7	34	9	23	10	76.2	61.3	115.3	186.5	19	16.2	23.8	.6					

I.H.P. during run..... 1,764.4
Pounds coal per I.H.P. per hour, as fired..... 1.93
Pounds coal per I.H.P. per hour, dry..... 1.85

Auxiliaries in operation, air pump, feed pump, stoker engine. Jet in smoke pipe.

NOTE.—Safety valve lifted slightly first two hours, then shifted cut-off on H.P. engine ¼ inch further out.

During the trial a test was made to ascertain the amount of water used by the stoker engines. To this end the exhaust from one of these engines was disconnected from the others and was led to a barrel containing cold water and weighed as it condensed. It was found that this engine used 35 pounds of water at 94 revolutions per minute, this being the average speed of the stoker engines under ordinary conditions. The cost, therefore, of four of these engines when running at the usual speed would be about 140 pounds of water per hour.

When running at the higher powers, on October 11 and 15, these engines averaged 125 revolutions per minute.

An approximate analysis of the coal used gave, as follows:

	Per cent.
Fixed carbon,	57.00
Volatile matter,	37.00
Moisture,	2.00
Ash,	4.00
	100.00

Heating value of coal by calorimeter, . . 13,580 B.T.U.

CORROSION OF BOILER TUBES IN THE UNITED STATES NAVY.

By Lieutenant-Commander Walter F. Worthington, U. S. Navy.

(Continued.)

The samples referred to in Table No. II, in the August number of the JOURNAL, p. 596, were returned to the porcelain-lined tank, submerged in distilled water, the air started flowing again, and the pieces exposed for sixteen weeks to as nearly as possible exactly the same treatment as was described before. Just before they were put in the tank small pieces were cut from samples Nos. 1, 7, 9 and 14, and submitted to an expert who made photomicrographs showing the internal structure of the metals. Unfortunately these photographs were not very clear, so that they could not be reproduced in the JOURNAL. Another attempt will be made to get better results. Incidentally the small pieces cut out changed the weight and area of those samples, for which due corrections have been made in Table IV. Also when the other pieces were reweighed, on June 30th, discrepancies of ± 1 gramme occurred in the case of samples 10, 11, 13 and 16, due no doubt to the fact that the two weighings were done by different persons, one of whom neglected a fraction of a gramme while the other counted it as a whole number, thus introducing in these four cases an error in weight of about two-tenths of one per cent.

Upon the conclusion of the second sixteen weeks' exposure, the air was shut off and the samples were left for a week under water, as it happened to be inconvenient to weigh them at once, then removed, washed with a brush (but not scraped) and weighed. The water in the tank was then carefully tested by a chemist and found to contain no free acid or alkali.

Table No. IV shows the areas and weights at the beginning

and end of the second period. The samples have the same numbers as they had in Table II, to which the reader is referred for many particulars which it was considered unnecessary to repeat. For convenience of comparison, however, the loss of weight per square inch for the first sixteen weeks has been given again in the last column of Table IV.

Two very important and positive results have been obtained by this second sixteen weeks' test. One is that the two 30-per cent. nickel-steel plates show most marked and unmistakable pitting. Much of the surface is as smooth as it was originally, while the rust concentrates at points, throwing up small craters of oxide encircling a deep pit.

The small loss of weight of these two samples, therefore, can not be taken as a measure of their resistance to corrosion. If this action continues the plates will soon be eaten through, and if they were boiler tubes they would be thereby rendered unserviceable. As the pieces are plates from which the mill scale has not been removed, they are not strictly comparable to seamless cold-drawn tubes, and it is possible that if the latter were made of 30-per cent. nickel they might do much better. Still the fact that this pitting is so marked ought to cause those who propose to use nickel-steel tubes to proceed with much caution.

The second important and positive result of the experiment is similar to the first, but applies to the seamless, cold-drawn, low-carbon, open-hearth steel tubes, made of the best and most homogeneous material possible, entirely freed from mill scale during the process of manufacture.

Samples 5 and 6, which in both the first sixteen weeks and also the second sixteen weeks were especially noticeable for their small loss of weight, have developed marked instances of the drilling action which is the most pernicious form in which corrosion can be manifested. Three deep, round holes, one to the left and below the Fig. 6 in sample No. 6, and two to the left and above the figure, are the most developed that appear on any of the sixteen pieces, while both samples 5 and 6 show a number of smaller round holes. The insides of these two samples have a smooth and almost unbroken surface.

TABLE IV.—SHOWING LOSS OF WEIGHT OF SAMPLES OF STEEL AND IRON AFTER THIRTY-TWO WEEKS' EXPOSURE TO AIR AND DISTILLED WATER.

Number of sample.	Kind of material.	Total area.	Weight at beginning of second sixteen weeks, June 30, 1900.	Weight at end of second sixteen weeks, October 20.	Total loss during second sixteen weeks.	Loss per square inch during second sixteen weeks.	Loss per square inch during first sixteen weeks.	Remarks.
		Sq. ins.	*Grammes.*	*Grammes.*	*Grammes.*	*Grammes.*	*Grammes.*	
1	Nickel-steel	55.4	255	251	4	.072	.053	Bent hot.
2	do	56.4	260	256	4	.071	.106	Bent cold.
3	Solid cold-drawn O. H. steel	40.88	387	377	10	.244	.342	These samples from one tube.
4	do	43.18	398	386	12	.277	.301	
5	do	53.62	508	500	8	.149	.149	These samples from one tube.
6	do	52.86	509	501	8	.151	.132	
7	do	56.57	440	424	16	.282	.298	These samples from one tube.
8	do	57.00	481	465	16	.281	.332	
9	Lap-welded Bessemer steel	56.99	455	439	16	.281	.244	All from one maker.
10	do	57.00	466	451	15	.263	.210	
11	do	57.00	504	488	16	.281	.228	
12	do	57.00	459	443	16	.281	.228	
13	Lap-welded iron	51.72	426	412	14	.271	.251	All from one maker.
14	do	55.83	448	433	15	.269	.241	
15	do	56.24	454	439	15	.267	.266	
16	do	57.06	474	458	16	.280	.262	

NOTE 1.—The weight and area are different in this table from Table No. III, because pieces were cut from samples Nos. 1, 7, 9 and 14 just before they were weighed and replaced in the tank, June 30, 1900.

NOTE 2.—In the case of samples 10, 11, 13 and 16, the discrepancy of ± 1 gramme between weights in tables III and IV is due to the fact that the weights were taken by different observers.

Samples 3 and 4, although produced by the same maker as samples 5 and 6, show a widely different character, the corrosion in this case spreading very evenly over the surface, the oxide adhering to the inside but leaving the outside clean and bright.

Samples 7 and 8, although produced by a different maker from samples 3, 4, 5 and 6, show corrosion spreading evenly over the surface like that of samples 3 and 4, and varying from it only a small amount.

The rate of corrosion of 7 and 8 is slightly less than it was for the first sixteen weeks, possibly due to the surface of the metal having been bruised by the process of drawing through the dies, rendering it more subject to corrosion at first until the outer skin had rusted and peeled off.

Samples 7 to 16, comprising seamless O. H. steel, lap-welded Bessemer, and lap-welded knobbled-hammered charcoal iron, show practically the same rate of corrosion, the rate having increased slightly in the case of the lap-welded steel and iron tubes, possibly due to the fact that the mill scale at first furnished a temporary protective covering.

In all of the samples except 1, 2, 5 and 6 the rust shows a tendency to stick on the inside of the piece, leaving that surface much rougher than the outside.

In samples 1 and 2 the inner and outer surfaces are much alike, while in samples 5 and 6 there is a considerable difference, the inside being far smoother than the outside, in each case.

Owing to the widely different manner in which the corrosion attacked samples 3 and 4 on the one hand and 5 and 6 on the other, a piece $\frac{3}{8}$ inch wide was shaved from the edge of sample 6 so as to include about a dozen small pit-marks, and the shavings analyzed. By comparing the analysis with that of sample No. 3 in Table II, it will be seen that the chemical composition of the metal in the tube represented by samples 3 and 4, is practically identical with that of the tube represented by samples 5 and 6.

The analyses are as follows:

	C.	P.	S.	M.	Si.
Sample No. 3, . .	.20	.010	.021	.40	.005
Sample No. 6, . .	.21	.010	.021	.40	.010

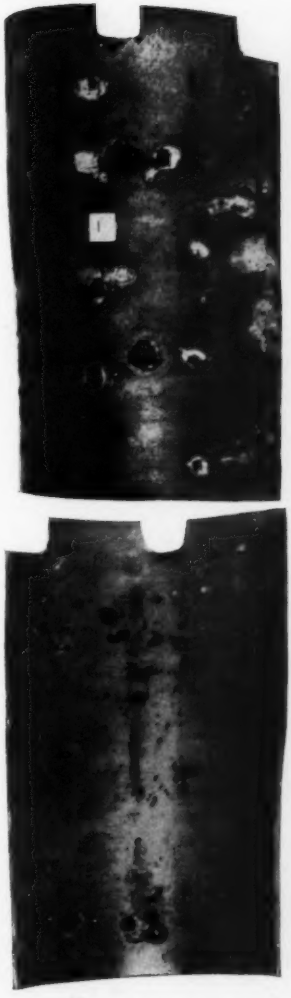


FIG. 1.

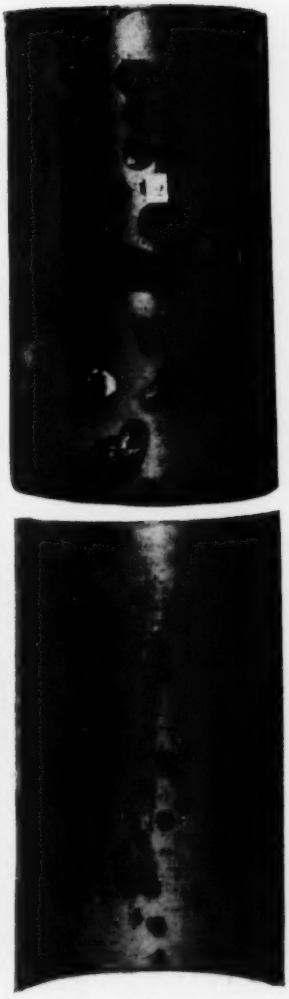


FIG. 2.

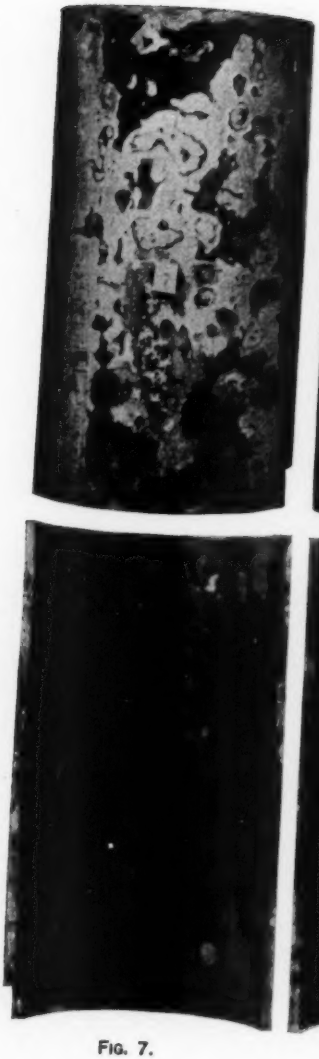


FIG. 7.

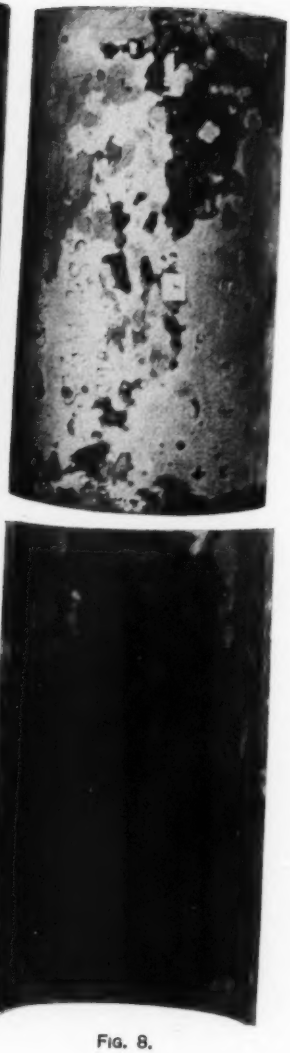


FIG. 8.

FIG. 9.

FIG. 10.

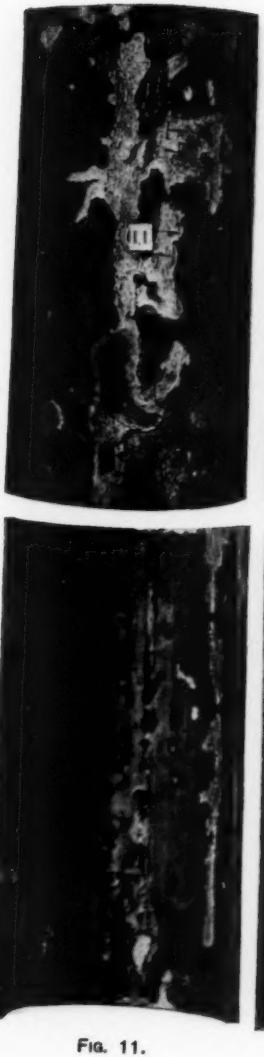


FIG. 11.

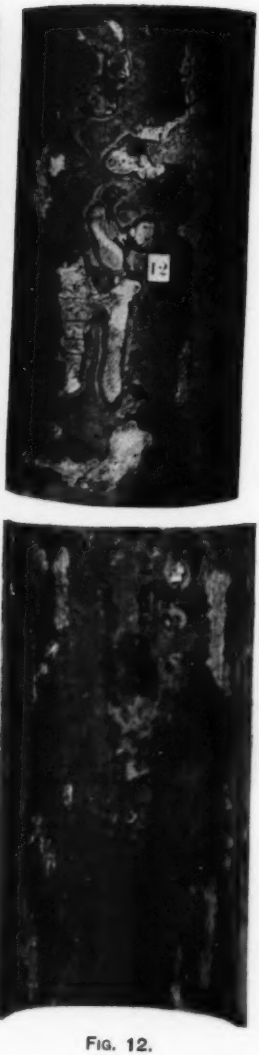


FIG. 12.

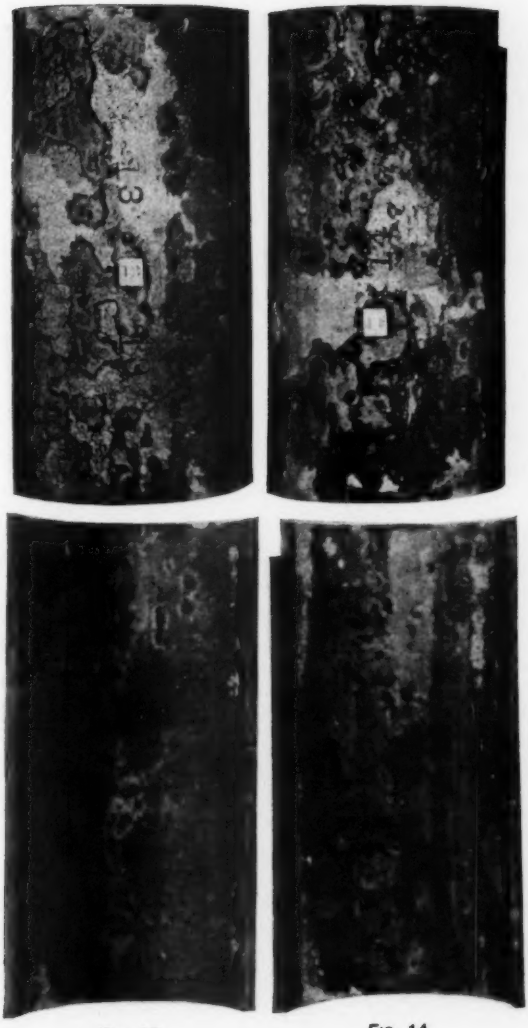


FIG. 13.

FIG. 14.

FIG. 15.

FIG. 16.

The shavings were cut in a very small shaping machine. Both the sound of the tool when cutting and the frequent slipping of the belt indicated that the metal was not of uniform hardness, but showed hard and soft spots. This is not to be wondered at when the method of manufacture of the tube is considered. A round billet of suitable length is first pierced by a very rude machine which leaves the inner and outer surfaces exceedingly rough. Subsequent hot and cold processes to which the hollow billet is subjected do not shave off the high places but just press them down into the body of the material. No amount of annealing can eliminate this inequality in the density of the metal. As the tubes go through the series of drawing processes to reduce them to the required dimensions, many of them are found to have fine, hard cracks on the surface at right angles to the axis, undoubtedly due to hard spots which lack the necessary ductility. Of course all tubes with such cracks on the outer surface are rejected, but if the cracks occur on the inside they cannot be seen.

The writer does not wish to be understood as asserting that the pitting is due to the inequality in the density of the metal in different parts of one tube or a corresponding difference in different tubes (since all billets are not left equally rough), but only wishes to point out the fact that such difference exists, and therefore it is desirable that researches be made in this direction. For example, if a lot of fifty or a hundred tubes were made from billets which had been drilled and turned instead of pierced, and if this lot, together with an equal number of tubes made by the ordinary method, were subjected to corrosion tests on the same plan as the sixteen samples referred to in this article, it would soon be seen whether or not drilling and turning the billets would remove the only defect known in cold-drawn seamless tubes as now manufactured in this country.

While the cost of drilling and turning the billets would be much greater than the cost of piercing them as now practiced, this extra expense would be largely offset by the saving of the great number of billets which are split and destroyed in the piercer, and also the saving of a large per cent. of such tubes as

are now rejected when they go through the drawing benches. In any case tubes made from drilled and turned billets would be cheaper than brass tubes formerly used in boilers, and also cheaper than nickel-steel of 20 to 25 per cent. nickel.

The accompanying photographs show the appearance of the inside and outside surfaces of the tubes at the end of the second period of sixteen weeks' exposure. The pieces are placed with the same end up as when photographed the first time. By comparing these photographs with those previously published it can be seen that in most cases the corrosion has attacked the same places as before, the area attacked being deepened and spread out, and this notwithstanding that in the interval between the two sets of experiments the samples were kept out of the tank for two and a half months.

Sample No. 3 shows on the outside some curious, small ellipses, semi-ellipses and closed curves with broken lines each, as if traced with a scriber, and plainly visible in the photographs.

Samples 7 and 8, which were left almost smooth by the peeling off of the first thin oxide, now appear much rougher from the formation of more rust.

INERTIA STRESS OF ELASTIC GEARS.

By J. H. Macalpine, Esq., Associate Member.

[Concluded.]

Chapter XII.

U. S. S. *CHICAGO'S* VALVE-GEAR. EFFECT OF STIFFENING GEARS. SIMILAR GEARS.

76. CHANGE IN *CHICAGO'S* GEAR—CONSEQUENT 1ST PERIOD MOVEMENT.

In Fig. 31 is shown an exceedingly simple modification of the Marshall gear. This diagram is so like Fig. 26 that it need not be explained. The center L of the reversing lever OL is merely moved a short distance from YY_1. A moment's consideration will show that if in making the change from the ordinary setting in which L lies in YY_1, we have to rotate SL, about S, in the same direction as the reversing lever OL rotates in passing from astern to ahead gear, the full ahead valve motion will be got with a smaller angle, OLK, of the reversing lever to LK than in the ordinary setting, LK being normal to SL. That is, the valve is slightly in ahead gear when the reversing lever lies along LK; and this will also appear from the equations of 1st period motion.

Indeed, a full ahead motion would be obtained with this normal position of the reversing lever if the angle LSY_1 were made large enough; but then no astern motion would be possible. A small angle displacement, SL, however, not only greatly improves the ahead motion, but leaves that for the astern perfectly practicable.

The advantages of this slight change are so obvious that I had always used it, with good results, in the few engines I had designed with the Marshall gear. Commodore (now Admiral)

Melville decided to adopt it in the new engines of the *Chicago*. As the engines were nearly completed when the modification was adopted the angle SLY_1 had, for certain clearances, to be made rather larger than I had previously used or consider necessary; and the angular travel of the reversing lever was also limited. Thus, in this case, the valve diagram, Fig. 32, shows larger leads than would be adopted in most cases. The results, however, have been entirely satisfactory.

With reference to this gear and an alteration in the gear of the *Petrel*, carried out about the same time and referred to below, Admiral Melville writes me under date of 18th December, 1899:

"With reference to the altered valve gear of the *Chicago:*

"It works splendidly, as also does the valve on the *Petrel*. There is a great improvement, both in the handling of the engines and economy."

The last sentence, of course, applies to the *Petrel*.

The greater smoothness of those valve curves over the *Newark's*, Fig. 29, is evident at a glance.

Putting $LSY_1 = \xi$, and $\beta =$ the angle by which the eccentric is in advance of the crank, we have—

1st period movements of point C,

(a) Parallel to SL. $\quad a\cos(\theta + \beta - \xi)$.
(b) Perpendicular to SL. $\quad -(m-1)\,a\sin(\theta + \beta - \xi)$.
(c) Perpendicular to SL. $\quad R_1Q_1\cos(\theta + \beta - \xi)$.

The first, (a), is evident from Fig. 31; while (b) and (c) are the same as given in § 59.

Only the resolved parts of these movements parallel to XX_1 will sensibly affect the valve, if the link CD is reasonably long. Hence we have—

Movement of valve, from (a) $= \quad a\cos(\theta + \beta - \xi)\sin\xi$.
Movement of valve, from (b) $= -(m-1)\,a\sin(\theta + \beta - \xi)\cos\xi$.
Movement of valve, from (c) $= \quad R_1Q_1\cos(\theta + \beta - \xi)\cos\xi$.

If the reversing lever lies along KL, $Q_1 = 0$—see equation (14); but it is readily seen that (a) and (b) alone give the valve an ahead movement, as stated above.

Fig 31.

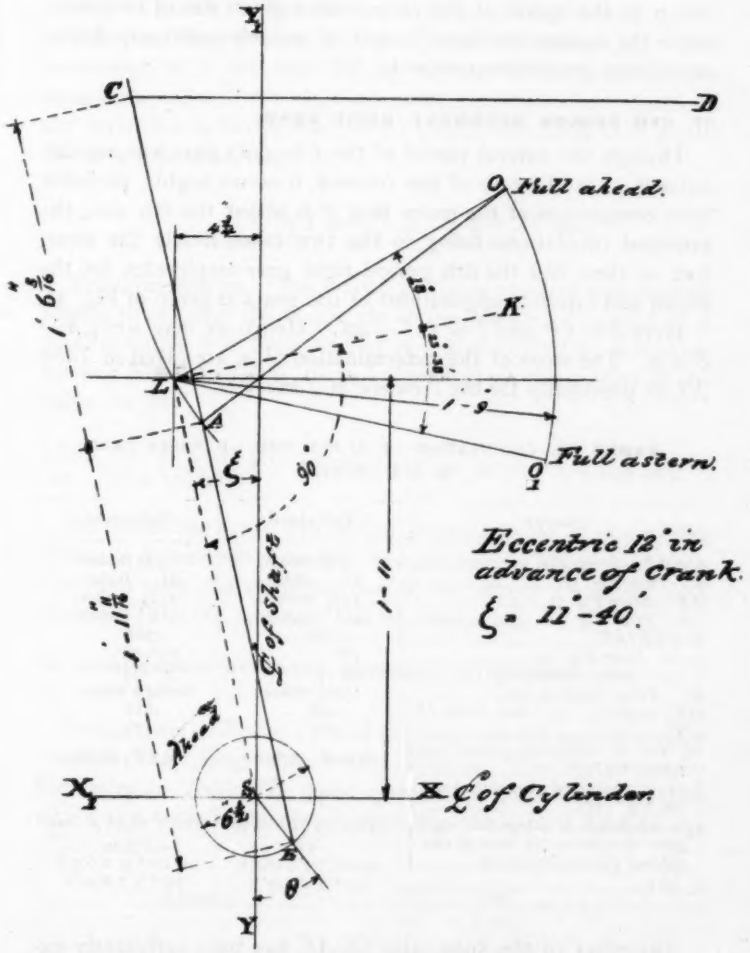


Diagram of L.P. Valve Gear.

U.S.S Chicago.

The amplitude of (a) is $a \sin \xi$, which is quite a sensible addition to the travel of the valve from a given size of eccentric; while the movements from (b) and (c) scarcely suffer any diminution from projection parallel to XX.

77. 6TH PERIOD MOVEMENT, RIGID GEAR.

Though the natural period of the *Chicago's* gear was not calculated, as in the case of the *Newark*, it seems highly probable from comparison of the gears that it is about the 6th also, the intended revolutions being, in the two cases, nearly the same. Let us then find the 6th period rigid gear amplitudes for the ahead and astern configurations of the gears as given in Fig. 31.

Here $\beta = 12°$ and $\xi = 11° - 40'$. Hence we may write $\beta - \xi = 0$. The steps of the determination of a_6 are given in *Table XI*, as previously for the *Newark* in *Table VII*.

TABLE XI.—CALCULATION OF a_6 FOR THE L.P. VALVE GEAR, U. S. S. *CHICAGO*.

Quantity.	Full ahead.	Full astern.
$a = SE$. From Fig. 31	3.25 inches.	3.25 inches.
R. From Fig. 31	21 inches.	21 inches.
AE. From Fig. 31	$23\frac{11}{16}$ inches.	$23\frac{11}{16}$ inches.
CE. From Fig. 31	$41\frac{1}{2}$ inches.	$41\frac{1}{2}$ inches.
$m = CE/AE$	1.768	1.768
$\psi - \xi$. From Fig. 31	18°	23°–38′
$\psi_1 - \xi$. From equation (94)	29°–52′	37°–43′
R_1. From equation (95)	15.67 inches.	18.45 inches.
$k/R_1 = \sin(\psi_1 - \xi)$. See *Table II*	.498	.612
a/R_1	.207	.176
1st term of substituted geometrical series, Fig. 24	.750 R_1 inches.	.857 R_1 inches.
Ratio of substituted geometrical series, Fig. (25)	.199	.224
a_6 = amplitude of 6th period rigid gear vibration = 7th term of substituted geometrical series	$\frac{.750 \times 15.67 \times .199^6}{12}$ $= 10^{-5} \times 6.082$ ft.	$\frac{.857 \times 18.45 \times .224^6}{12}$ $= 10^{-4} \times 1.665$ ft.
$a_6 \cos \xi$	$10^{-5} \times 5.957$ ft.	$10^{-4} \times 1.630$ ft.

The effect of the finite ratio SE/AE has been sufficiently examined for the case of the *Newark* as given in Fig. 26. But in Fig. 31, any irregularity of the movement of A will by projection be directly transmitted to the valve motion. Hence it may be

suspected that in this way the amplitude of a_6 may be seriously raised. That this is not the case may be readily shown.

Take the simple case in which A moves along SL. The movement of A will then be represented by an equation like equations (10) and (11). In the papers from which these equations are derived it is shown that the 6th period amplitude of A would be, very approximately,

$$\frac{a}{12} \times \frac{1}{512} \left(\frac{a}{AE}\right)^5 = \frac{3.25}{12} \times \frac{1}{512} \left(\frac{3.25}{23\frac{11}{16}}\right)^5 \text{ feet.}$$

The investigation of the motion of the connecting rod in those papers shows that the amplitude of the 6th period motion of C parallel to SL is m times that of A. Multiplying by m and resolving parallel to XX_1, to find the effect on the motion of the valve, we get

$$\frac{3.25}{12} \times \frac{1}{512} \left(\frac{3.25}{23.6875}\right)^5 \times 1.768 \sin 11^\circ 40' = 10^{-9} \times 9.194 \text{ feet.}$$

No doubt the actual 6th period irregularity of A, due to this cause, is somewhat—possibly several times—larger than this, but even then the amplitude would be minute compared with those in *Table XI*, and need not be considered.

78. COMPARISON OF VALVE MOTIONS OF *CHICAGO* AND *NEWARK*.

Comparison of *Tables VII* and *XI* show a much more favorable result for the *Chicago* than for the *Newark*. The best result for the latter in *Table VII* gives $a_6 = 10^{-4} \times 4.45$, and the ratios which this bears to the ahead and astern values of $a_6 \cos \xi$ for the *Chicago* are—

$$\text{Ahead, ... } \frac{10^{-4} \times 4.45}{10^{-5} \times 5.957} = 7.47.$$

$$\text{Astern, ... } \frac{10^{-4} \times 4.45}{10^{-4} \times 1.630} = 2.73.$$

From the results we can at once anticipate that the *Chicago's*

gear will work with great smoothness, especially when running ahead, but also when running astern.

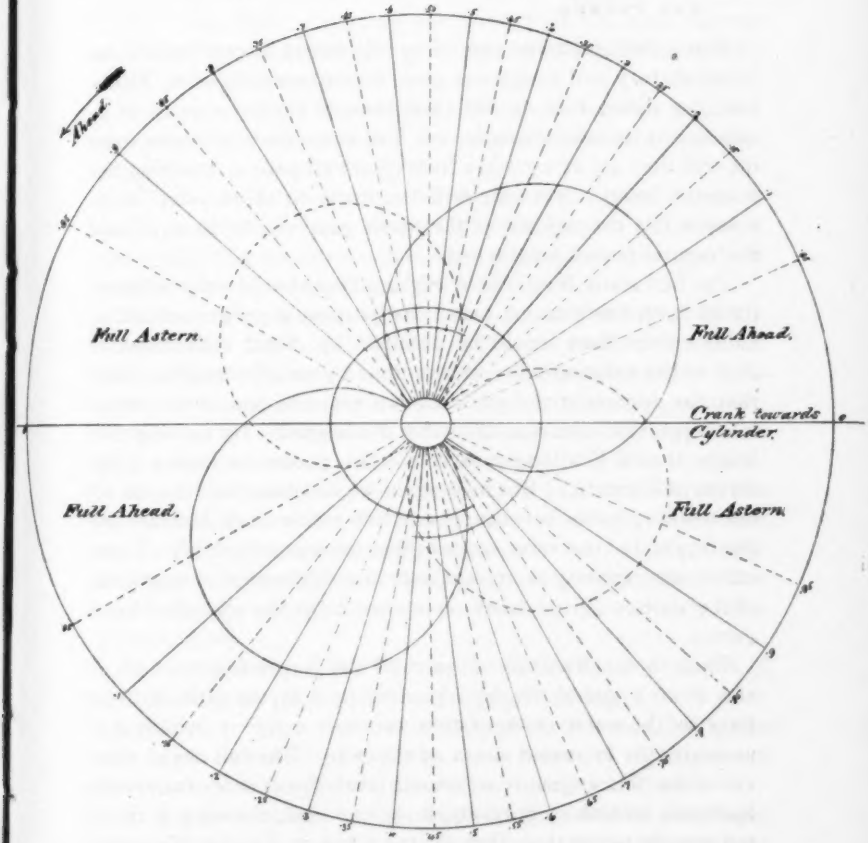
The ratios of the geometrical series are smaller in *Table XI* than in *Table VII.* These are the ratios of the substituted geometrical series, the true ratios being Q_2/Q_1, Q_3/Q_2, &c. From Fig. 23 we can at once see that, for the values of k/R_1 and a/R_1, in the above tables, the true ratios will not differ by a large percentage from the substituted ones. Hence, all the amplitudes a_2, a_3, &c., in the full ahead valve motion of the *Chicago* will be much smaller than for even the .65 cut off in the *Newark's* gear. The same is true to a lesser extent in the full astern valve motion of the *Chicago.* The increase of smoothness of the curves in Fig. 32 over those in Fig. 29, which follows from the diminished amplitudes, is markedly evident to the eye.

Examining the stresses from Fig. 32 in the same way as is done for the *Newark* in *Examples I* and *II*, § 10, we find for the ahead curves that the maximum occurs very near the intersection of the dotted radial line marked .15 in the lower half of the crank circle; that is, almost exactly at the end of the stroke. The stress here only exceeds by 20 per cent. the maximum we would get if the motion was strictly simple harmonic. For the astern valve motion the maximum stress occurs at the point of intersection of the dotted radial line marked .95 in the lower half of the crank circle, and the excess over simple harmonic motion with the astern travel is rather under 45 per cent. In *Example II*, § 10, for the *Newark* the estimated excess was 86 per cent.

It is also interesting to compare the angle of crank between the maximum displacements of the valve to right and left in Figs. 29 and 32. If the motion was strictly simple harmonic a maximum displacement would occur every half revolution; that is, for 180° crank angle. In Fig. 29 this is departed from by the difference of the angles $E°$ and $F°$; and in *Table X* we find $E° - F° = 14°$. In Fig. 32 it will be found that this difference is reduced to 9°.

All these results flow from the diminished amplitudes of the

Fig. 32.



L.P. Valve Diagram.
U.S.S. Chicago.

simple harmonic motions of successive periods which go to compose the whole valve motion.

79. REMARKS ON VARIOUS GEARS, U. S. S. *KATAHDIN* AND *PETREL*.

Any considerable increase of m will nearly always lead to an unsatisfactory and dangerous gear, for two reasons,—1st. Equation (94) shows that we will then have an excessive value of ψ_1 unless ψ is unusually small. 2d. The stress from the valve spindle will then act at a greater leverage on all parts controlling the eccentric beam. Not only will the stress on those parts be increased but the rigidity of the whole gear will be lowered and the natural period lengthened.

The bell crank lever VGH, Fig. 26, also, should only be introduced with the greatest care. Its purpose is to get smoother valve curves than would be obtained by direct connection of A_1V to the valve spindle; this is done by so adjusting the lever that the portion of the valve motion near one end of the travel is exaggerated, and near the other diminished. By making GH larger than GV a longer valve travel is produced from a given throw of eccentric. But here again we increase the stress on all the working parts between V and the crank shaft, increase the elastic yield of the valve and lengthen the natural period. These effects may entirely counterbalance the diminution of amplitude of the shorter period harmonics, shown by the smoother valve curves.

Thus in the *Katahdin*, if we draw the perpendiculars from H and V on a line through G parallel to XX_1, the ratio of these lines in the worst configuration becomes 1.776 : 1, which also measures the increased stress on the gear. The full ahead valve curve for this engine is so smooth that the excess over simple harmonic motion is only about 30 per cent., showing a result not greatly worse than that for the *Chicago*, Fig. 32. Nevertheless the valve gears vibrated as shown by the following quotations from Official Reports:

Extract from the Report of the Trial Board of U. S. S. *Katahdin*, dated Nov. 2d, 1895.

"A moderate vibration was noticeable in the valve gear of both engines. The I.P. and L.P. valve gears in both engines was linked up to its minimum cut off to reduce the strain on it."

Extracts from the reports of the Board of Inspection.

April 6th, 1896. "During the whole trial, the main engines worked well, except the valve gear of the starboard I.P. and L.P., which vibrated considerably."

April 10th, 1896. "The engines worked sitisfactorily, with the exception of vibration in low-pressure and intermediate valve gears, which was excessive."

In U. S. S. *Petrel*, also, the curves showed only an excess of about 30 per cent. above simple harmonic motion. This result was obtained by the introduction of several links and levers, which greatly reduced the rigidity of the whole gear.

On trial this gear also vibrated badly. In the Report of the Forced-Draft Trial of July 20th, 1889 (2d trial), it is stated: "Wooden chocks were placed between the frames and the ends of the reversing rock-shaft arms of each engine to prevent the vibration of the valve gear." From other parts of the Report the prevention does not seem to have been anything like complete.

About the same time as the *Chicago's* gear was altered, the valves and gear of the *Petrel* were also modified. The valve travel, which was unnecessarily long, was reduced, thus not only raising the mean stiffness of the gear but diminishing the effects of the angular swing of the rods. The result has already been noted in § 76.

In such gears as Joy's, Morton's or Kirk's the smoothness of the resulting curves depends on the proper setting of those parts introduced to effect the compensation of the motion, but great care must be exercised that the same defects of rigidity as we have noticed in the *Katahdin* and *Petrel* is not introduced.

In the gear treated by Zeuner under the name of Hensinger von Waldegg—also known as Walschaert gear—an excellent valve motion may be attained if the various links are kept rasonably long without sacrificing rigidity.

In smoothness of valve motion the original Hackworth gear with its straight slide, or still better with a slide curved to a

properly determined radius, must be greatly superior to the ordinary setting known as the Marshall gear; though the slide of the former is mechanically inferior to the link in the latter, especially if the surfaces are not very ample.

80. EFFECT OF STIFFENING GEARS. GENERAL CONSIDERATIONS.

The inquiry naturally arises from what immediately precedes, What is the effect on the maximum stress, of stiffening the gear?

In the two gears referred to in *Chapter I*, which first directed my attention to the present subject, though it was not practicable to alter the valve motion it was possible to stiffen the gear. This, as we have seen, was also done in the *Petrel's* engine by fitting wooden chocks so as to eleminate all the elasticity of the reversing gear. Let us examine what are the exact circumstances under which the maximum inertia stress, I_n, from the actual synchronizing vibration will be reduced. The magnification of amplitude of other periods will also be somewhat altered, but the change in I, will usually be most important.

A gear in which there is synchronism at full speed might, by comparatively slight stiffening, have the natural period altered so that this was avoided and a very obvious improvement in the result would be produced. On the other hand, slight increase in rigidity of a gear in which the natural period was non-synchronous might produce synchronism and greatly increased stress and unsteadiness in the running of the gear. As an example, suppose the *Newark's* natural period raised so that $s = 6.5$, the magnification of the 6th and 7th period amplitudes will then be—

$$\left.\begin{array}{ll} \text{6th period.} & \dfrac{6.5^2}{6.5^2 - 6^2} = \quad 6.760 \\ \text{7th period.} & \dfrac{6.5^2}{6.5^2 - 7^2} = -6.259 \end{array}\right\} \text{See equation (101).}$$

Both these values are very much lower than those in line (oo), *Table X*, where exact synchronism is supposed.

Obviously a change of about 10 revolutions either way would

produce synchronism in this case, since the *Newark* has been taken as running 127 revolutions. So that the result just given is much better than would actually occur, as there will certainly be a fluctuation of a few revolutions up of down which would being the 6th or 7th period much more nearly into synchronism with the natural period and produce a much greater magnification of amplitude of the corresponding vibration. If we had been dealing with higher periods than the 6th or 7th a still smaller change in the number of revolutions would have produced synchronism. The corresponding effects are well known in connection with the vibrations of steamships where a slight change in the pitch of the propeller, altering correspondingly the full power revolutions of the engines, has often relieved a ship from objectionable vibrations.

81. SOLUTION WHEN STIFFENING RAISES SYNCHRONIZING TERM FROM s TO $s+1$.

The important problem then is to examine the effect of stiffening the gear where both before and after stiffening there is exact synchronism; but, of course, not of the same period.

Let the stiffening be such that the natural period is changed from s to $s+1$.

Let the quantities a_s, ω_s, I_s, τ, a, f,
be changed to a_{s+1}, ω_{s+1}, I_{s+1}, τ_1, a_1, f_1

The result may be obtained by the use either of equation (90) or (91_2). Using the former, we have, since F does not change—

$$F = \frac{\pi f s a_s}{\tau} \cos a = \frac{\pi f_1 (s+1) a_{s+1}}{\tau_1} \cos a_1.$$

Whence
$$\frac{a_{s+1}}{a_s} = \frac{\tau_1 \cos a}{\tau \cos a_1} \times \frac{fs}{f_1 (s+1)}. \quad . \quad . \quad . \quad . \quad (102)$$

$$\frac{s^2}{(s+1)^2} = \frac{\omega_s^2}{\omega_{s+1}^2} = \frac{f}{f_1}, \text{ by equation (17).}$$

Hence equation (102) becomes

$$\frac{a_{s+1}}{a_s} = \frac{\tau_1 \cos a}{\tau \cos a_1} \cdot \frac{s^3}{(s+1)^3}. \quad . \quad . \quad . \quad . \quad (103)$$

Now, by stiffening the gear we have made no change in I_e and I_f; and, thus, if we are to have

$$I_s = I_{s+1}$$

we have still to deal with the same curves n_e and n_f, of energy transfer and change of epoch, in determining τ_1 and α_1, as those from which τ and α were determined. By determining the angles $l'Oe$ and $r'Of$, as in *Table X*, and from those determining the normal ordinate and ordinate of the n_f curves which must be superimposed on the $O°$ ordinate of the n_e curves; and also by describing the line curve *EFGHK*, as in Fig. 30_1, &c., we can measure as before τ_1 and α_1.

For instance, with $n_e = 2.0$ and $n_f = 3.3$ but using $s + 1 = 7$ instead of $s = 6$ we readily find from *Table X* that the energy transfer curve *aaaa*, Fig. 30, and the curve of change of epoch due to friction, *aa*, Fig. 30_1, remain almost unchanged. (Observe footnote to line (s), *Table X*.)

The normal epoch changes to 88°. The amplitude of the sine curve *EFGHK* is, by equation (85), reduced in the proportion

$$\frac{f_1(s+1)\,a_{s+1}}{fsa_s} = \frac{(s+1)^3\,a_{s+1}}{s^3 a_s} = \frac{7^3}{6^3} \times .305 = .484,$$

since a_{s+1}/a_s = the ratio of the substituted geometrical series, .76 cut off, *Table VII*. This leads to values of α_1 and τ_1 little different from the correct values of α and τ. (It will be remembered that the correct value of α, .76 cut off, is considerably less than that due to the singular point α, Fig. 30.) Hence the ratio

$$\frac{\tau_1 \cos \alpha}{\tau \cos \alpha_1}$$

in equation (103) is very near unity.

The same conclusion will be drawn from examination of the cases in Figs. 30_2 and 30_3.

In a case like that to which *dddd*, Figs. 30 and 30_4, refer where a small change in the quantities involved may make τ_1 very different from τ, the above ratio may not be near unity.

To conclude:

1st. Equation (103) gives the exact ratio a_{s+1}/a_s which must hold in all cases if I_{s+1} is to be equal to I_s. If the actual ratio a_{s+1}/a_s is smaller than that given by the right-hand side of this equation, the gear will be improved by stiffening.

2d. In many cases—probably by far the largest number of cases—the ratio

$$\frac{\tau_1 \cos a}{\tau \cos a_1}$$

is very near unity, and equation (103) becomes very approximately,

$$\frac{a_{s+1}}{a_s} = \frac{s^3}{(s+1)^3}.$$

If $s = 6$ we then have

$$\frac{a_{s+1}}{a_s} = \left(\frac{6}{7}\right)^3 = .630$$

If s is greater than 6, this ratio would also be greater.

The largest ratios for the *Newark* and *Chicago* were—

Newark, Table VII. Ratio = .305.
Chicago, Table XI. Ratio = .224.

3d. These ratios show that in any practical case

$$\frac{\tau_1 \cos a}{\tau \cos a_1}$$

will require to be much smaller than unity if the valve gear is not to be improved by stiffening. But this may very readily happen if the actual epoch of the synchronizing vibration, E_s', falls at a point of the energy transfer curve where τ is small and rapidly varying with change of epoch.

Equation (103) will always make the prediction of such a case possible; and if it is the gear of an engine already built and tried, s may be found, by observation, so that the tedious part of the process—calculating the natural period—may be avoided.

82. SIMILAR GEARS HAVING s AND F/I_e THE SAME WILL BE SIMILARY STRESSED.

Similar valve motions.

There is a simple proposition with regard to similar valve motions with similar stresses, which is sufficiently interesting to be included here.

Proposition. If the rigid gear valve motion from two gears are similar; and, also, if s and F/I_e are the same for both, the actual valve motions and stresses will be similar.

The equations of motion, equation (1), for the two gears will only differ in the amplitudes $a_1, a_2, \ldots a_n$, which will be in proportion to the linear dimensions of the gears.

As s remains the same so also will $T/T_1, T/T_2, \ldots T/T_n$; hence the actual valve motions due to the nonsynchronizing terms remain similar.

$$I_e = \frac{M\omega^2}{g} r_e \left(\frac{r_e}{\rho_e} - 1\right) \text{ from line (m), } \textit{Table X.}$$

$s = \frac{1}{\omega}\sqrt{\frac{fg}{M}}$, equation (99), and the proportionality of all the linear dimensions $a_1, r_e, \rho_e, a_n, \ldots$ allow us to write this

$$I_e = const\, f a_1. \quad \ldots \ldots \quad (103_1)$$

The ratio n_e/n_f remains unaltered; and also all the angles $E°, F°, l'Oe, r'Of, \ldots$ of Fig. 29, are the same for the two gears. Hence for each value of n_e, the diagrams of energy absorption and change of epoch through friction and the normal epoch are the same. Thus the conditions which would give rise to the same value of α in the two gears would have τ and κ, and consequently also, $\cos \alpha/\tau$ and $\sin \alpha/\tau$, unchanged. Equation (90) or (91_2) then give

$$F = const\, f a_e = const\, f a_1;$$

which makes F proportional to I_e, by equation (103_1).

As n_e is unaltered $I_e = n_e I_e$ is also proportional to I_e; that is, all three elements of stress change in the same proportion in passing from one gear to the other. As the epochs of the various vibrations all remain unchanged, the total stress is also changed in the same proportion.

CHAPTER XIII.

FRICTION PROPORTIONAL TO THE VELOCITY. VISCOSITY

83. FRICTION PROPORTIONAL TO THE VELOCITY.

In this chapter I shall treat two cases,

1st. In which the frictional resistance to the motion of the valve varies as the first power of the velocity.

2d. In which viscosity of the metal of the gear absorbs the energy of vibration of the valve.

The first case is of interest but much less importance than that treated in previous chapters; hence its treatment is brief.

The second case is of the greatest practical importance.

Friction proportional to the first power of the velocity.

The rigid gear motion of the valve being given by equation (1), and V being the frictional resistance of the valve in pounds when its velocity is one foot per second, it can be shown—see Note D, equation (22d)—that the actual motion of the valve is given by

$$r_1 = k + \Sigma_1^{\infty} \frac{a_n \omega_s^2}{\sqrt{\{(\omega_s^2 - n^2\omega^2)^2 + \mu^2 n^2 \omega^2\}}} \cos(n\omega t - \varepsilon_n - \nu_n)$$

$$+ Ae^{-\frac{\mu t}{2}} \cos\left\{\left(\omega_s^2 - \frac{\mu^2}{4}\right)^{\frac{1}{2}} t - \lambda\right\}, \quad . \quad . \quad . \quad (104)$$

where $$\mu = \frac{Vg}{M},$$

and $$\tan \nu_n = \frac{\mu n \omega}{\omega_s^2 - n^2\omega^2}. \quad . \quad . \quad . \quad . \quad . \quad (105)$$

A and λ are arbitrary constants, as before, and e is the base of Nepierian logarithms.

The factor $e^{-\frac{\mu t}{2}}$ gives the law of extinction of any arbitrary disturbance.

If μ is small and $n\omega$ much smaller than ω_s (T/T_n small) ν_n is a small positive angle. That is, the actual vibration of the nth period then lags slightly behind the corresponding rigid gear vibration. Equation (105) also shows that as we pass to larger values of n the angle ν_n increases.

When there is synchronism ($n = s$) the denominator of equation (105) vanishes—equation (22_1)—and

$$\nu_s = \frac{\pi}{2};$$

that is, the actual synchronizing vibration lags one quarter period behind the rigid gear vibration, as in the case of no resistance.

The amplitude of the actual vibration

$$\frac{a_n \omega_s^2}{\sqrt{\{(\omega_s^2 - n^2\omega^2)^2 + \mu^2 n^2 \omega^2\}}} \quad . \quad . \quad . \quad . \quad (106)$$

then becomes $$a_s' = \frac{a_s \omega_s}{\mu}.$$

If the friction is small a_s'/a_s is large; but not infinite, as in the case of unresisted motion.

Differentiating expression (106) with respect to n, we readily find that the maximum magnification of amplitude takes place for a lower value of n than would give synchronism. At the maximum amplitude we have

$$\omega_s^2 - n^2\omega^2 = \frac{\mu^2}{2} \text{ and } tan\, \nu_n = \frac{2n\omega}{\mu}.$$

Then $$\frac{a_n'}{a_n} = \frac{\omega_s^2}{\mu\sqrt{\left(\omega_s^2 - \frac{\mu^2}{4}\right)}}.$$

The reason of this result is readily seen. With a difference of phase between the actual and rigid gear vibrations of rather less than a right angle, they will be practically as well placed for transmitting energy to the valve as when the difference is exactly a right angle. But if we compare a vibration of the nth period, n being rather smaller than s, with the synchronizing vibration, we see that for a given amplitude of the actual vibration the velocity at each part of the cycle will be smaller for the former than for the latter. Hence the energy absorbed per cycle will be smaller; and—supposing the same rigid gear amplitudes in the two cases—if in the latter the energies communicated and absorbed are equal, so that the amplitude is constant, in the

former case there will be an excess of energy communicated which will cause the amplitude to grow.

84. NATURE OF CONDITIONS AND STRESSES IN A MACHINE. EFFECT ON COEFFICIENT OF VISCOSITY.

Viscosity.

The material of the working gear and frame of any machine is subjected to the periodic recurrence of stress; but according to a more complex law than the simple harmonic torsional vibrations of the wires in Tomlinson's and Wiedemann's experiments, referred to in *Chapter IV.* In the latter case, also, it was found that the conditions as to temperature and constancy in the nature of the recurring strain had to be very uniform to allow the coefficient of viscosity to fall to a minimum.

In the case of any gear we may assume the constancy of conditions which leads to the minimum viscosity; for this is the worst case, and it will very frequently occur. It would not be justifiable to suppose that fluctuations of 2° Centigrade, or less, in the temperature of the air, which Tomlinson found to be the cause of the viscosity in *Experiment VII, Table IV* being more than double that in *Experiment III,* is going to affect the comparatively large masses of metal in the gear to the same extent as it did the fine wires of the experiments. In the gear the change of temperature of the metal—which is obviously the real cause of the rise of viscosity—will be effected very many times more slowly than in the fine wire, and "accommodation" will keep pace more or less nearly with this change.

If the machine is running uniformly, the nature of the recurring strain to which each part is subjected—be it extension and compression, torsion, or transverse strain, or any combination of these—remains exactly the same in succeeding revolutions. Here also, then, we may expect a definite minimum of viscosity, unless the limits of strain are so wide that fatigue of elasticity presents itself; but this, possibly, would only precede break down of the gear. I am not aware that the limits beyond which we get fatigue of elasticity have been investigated, but we know that they are wide.

Let us take the case if any gear, such as that of the *Newark* treated so fully above, but suppose that the synchronizing vibration has not grown so as sensibly to contribute to the maximum stress. We find a periodic variation of the stress—not exactly simple harmonic—such that there are two maxima, one near each end of the stroke. In this motion viscosity plays no sensible part; for, in any reasonably stiff gear, it differs very slightly from the rigid gear motion. How extremely slight its influence is here, we will be able to conclude immediately from an examination of relative absorption of energy by friction and viscosity in the 6th period vibration of the *Newark's* L.P. valves. For a great portion of the revolution, between the two maxima, the gear is submitted to a comparatively light stress.

Let us now superpose on this movement a synchronizing vibration of high period, and of such amplitude that the maximum stress due to it alone equals or exceeds the greatest of the above maxima. The gear is now submitted to a large number of maxima in each revolution and these are of very different intensities. The greatest is when a maximum from the synchronizing vibration falls very near the highest of the two maxima from the nonsynchronizing vibrations, the strains from them being additive. Usually the smallest maximum will practically coincide with the next preceding or following opposite maximum of the synchronizing vibration, when the strains from the nonsynchronizing and synchronizing vibrations oppose one another. In the case suppose this maximum may have any value from zero upward. The other maxima will be of an intensity intermediate to those two.

That this continual change of intensity of the maxima, occurring periodically each revolution, will raise the coefficient of viscosity, there does not seem any sufficient reason to conclude *a proiri*, and I know of no experimental evidence that throws light on the question.

As given in *Table IV*, the subsequent calculations are based on a rate of absorption of four-tenths per cent. of the total energy of vibration per complete vibration. This figure is arrived at from far too narrow a basis to make even its approximate

accuracy probable. But a large proportionate change in its amount would affect the nature of the results but little.

85. ABSORPTION OF ENERGY BY FRICTION AND VISCOSITY COMPARED.

As it will appear that viscosity has little influence unless a_n'/a_n is large, we need only deal with the synchronizing vibration.

As the energy of vibration with a_s'/a_s large is

$$\tfrac{1}{2} f a_s'^2,$$

the energy abstracted by viscosity each complete vibration is

$$.004 \times \tfrac{1}{2} f a_s'^2 = .002 f a_s'^2. \quad . \quad . \quad . \quad . \quad (107)$$

The energy absorbed by friction per revolution being

$$\tau F a_s',$$

we get

$$\frac{\text{Energy absorbed by viscosity.}}{\text{Energy absorbed by friction.}} = \frac{.002 f s a_s'}{\tau F}. \quad . \quad . \quad (108)$$

The values of this ratio for the *Newark* are given in *Table X*, line (qq). These are so small that, even if the absorption by viscosity was raised to several times that here supposed, the calculation of stress in the preceding chapters would not be sensibly modified.

86. VALUE OF a_s'/a_s WHEN VISCOSITY ALONE ACTS.

If there is no friction and viscosity alone absorbs the energy which the gear continually transmits to the synchronizing vibration of the valve, we may readily calculate the magnification of amplitude which it will undergo.

By equation (88) the energy supplied by the gear per vibration is

$$\pi f a_s a_s' \cos \alpha.$$

Equating this to expression (107) we get,

$$\pi f a_s a_s' \cos \alpha = .002 f a_s'^2,$$

or

$$\frac{a_s'}{a_s} = 500\pi \cos \alpha = 1571 \cos \alpha. \quad . \quad . \quad . \quad (109)$$

There appears no reason whatever to conclude that the internal action in the metal will cause a to differ sensibly from zero; the energy absorbed per vibration is so small compared with the total energy of vibration that the effect must, at most, be feeble.* Instead of equation (109) we may thus write simply

$$\frac{a_s'}{a_s} = 1571. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (110)$$

The rigid gear synchronizing vibration will have to be exceedingly small if this large increase is not to bring it into prominence.

87. REQUIRED STIFFNESS OF GEAR IF VISCOSITY ALONE ACTS. *NEWARK* AND *CHICAGO*.

To bring out clearly the necessary limiting relation between the stiffness of any gear and the smoothness of motion which it imparts to the valve or other moving mass, if the movement is to be practicable without friction, let us suppose that instead of the actual rigid gear amplitudes $a_1, a_2, a_3, \ldots$, we have the substituted geometrical series $Q_1', Q_2', Q_3', \ldots$, ratio j. Then,

$$\frac{Q_s'}{Q_1'} = j^{s-1};$$

and, by equation (3),

$$\frac{\text{Stress due to } Q_s', \text{ rigid gear.}}{\text{Stress due to } Q_1', \text{ rigid gear.}} = j^{s-1}s^2.$$

As $Q_s' = a_s$ in any practicable gear, where Q_s' must be removed from Q_1' by a very considerable number of terms, we get by equation (110),

$$\frac{\text{Stress of actual synchronizing vibration.}}{\text{Stress of 1st period, due to } Q_1', \text{ rigid gear.}} = 1571 j^{s-12}s. \quad . \quad (111)$$

I will use this in two cases, one of which will sufficiently represent the results from both the *Newark's* and *Chicago's* valve settings given above.

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*If energy is abstracted from a nonsynchronizing vibration the epoch must increase to allow the energy to be restored by the gear.

TABLE XII.—TRUE RIGID GEAR STRESS OF U. S. S. *NEWARK* AND *CHICAGO'S* L.P. GEAR COMPARED WITH THAT FROM $Q_1' \cos \theta$.

Line.		*Newark.*			*Chicago.*	
(a)	Cut off	.76	.725	.65	Full ahead.	Full astern.
(b)	Q_0'. *Tables VII* and *XI*	1.030 R_1	.989 R_1	.924 R_1	.750 R_1	.857 R_1
(c)	R_1, in inches. *Tables VII* and *XI*	21.78	20.42	17.88	15.67	18.45
(d)	j. *Tables VII* and *XI*	.305	.289	.262	.199	.224
(e)	$Q_1' = Q_0' j$, in inches	6.842	5.837	4.329	2.339	3.542
(f)	Travel (=twice amplitude) of valve, in inches. Figs. 29 and 32	8.85	8.35	7.41	7.125	6.2
(g)	Excess of maximum rigid gear inertia over that due to simple harmonic motion. §§ 10 and 78	1.86	...	...	1.20	1.45
(h)	Amplitude which, with simple harmonic motion, would give true maximum rigid gear stress = line (f) × line (g) ÷ 2	8.23	...	...	4.275	4.495
(k)	True rigid gear stress / Stress from $Q_1' \cos \theta$. = line (h) / line (e)	1.203	...	...	1.828	1.269

Let us inquire, What would be the required value of j for a given stiffness of gear—that is, for a known value of s—if the actual synchronizing vibration is to grow till its inertia stress equals that from all the rest of the terms?

If its inertia stress is to equal that of the 1st period from Q_1' we have at once from equation (111),

$$1571 j^{s-1} s^2 = 1.$$

But *Table XII*, line (k), shows that, if the stress is to rise to equality with the actual rigid gear stress, it will have to be still further increased for the *Newark* full ahead by 1.203, and for the *Chicago* full ahead and full astern by 1.828 and 1.269 respectively. As fairly representing all these ratios we may take 1.5. The figures for the earlier cut offs in the *Newark* are not fully calculated, but probably 1.5 would nearly represent them also. We then have

$$1571 j^{s-1} s^2 = 1.5. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (112)$$

This may be supposed to apply equally to the case of the rigid gear stress and to the stress arising from the actual motion

due to the nonsynchronous terms, as they do not differ in a large proportion.

To find the effect of stiffening the gear we may also inquire what will be the relation of j and s if the stress from the actual synchronizing vibration is to be one-third of that in equation (112), or

$$1571 j^{s-1} s^2 = .5. \quad . \quad . \quad . \quad . \quad . \quad . \quad (113)$$

From equations (112) and (113) j can at once be determined for given values of s.

TABLE XIII.—CORRESPONDING VALUES OF j AND s FROM EQUATIONS (112) AND (113).

Equation $1571 j^{s-1} s^2 =$ s	(112) 1.5 j	(113) .5 j
6	.1215	.0976
7	.1640	.1366
8	.2044	.1747
9	.2421	.2110
10	.2768	.2450
11	.3088	.2767
12	.3382	.3061
13	.3653	.3334

88. EFFECT OF STIFFENING GEARS.

Table XII brings out clearly the great effect of stiffening the gear when viscosity is the only agent absorbing energy. Comparing the two columns of values of j we see that those in the left-hand column do not differ greatly from those one line lower in the right-hand column. Hence, by stiffening the gear so that s is raised by unity, the inertia stress from the actual synchronizing vibration is reduced to one-third of its value before stiffening.

If the rigidity of the gear was reduced from that in the left-hand column, so that s was decreased by unity, the stress from the actual sychronizing vibration would again be nearly trebled;

hence, for the various values of s, the ratios j in this column are about the greatest that would be in any way practicable, and even here the gear could not be considered to be running very smoothly.

Thus the *Newark* to run smoothly without any friction at .76 cut off, $j = .305$, would require to have its natural period raised from the 6th to the 11th; that is, to be stiffened $11^2/6^2 = 3.36$ times. Even at .65 cut off s could not be much below 10. The *Chicago* to run in full ahead gear, $j = .199$, would require $s = 8$; and full astern, $j = .224$, would require $s = 10$.

We also see that if the coefficient of viscosity, which we have taken as .004, was trebled, in the limiting conditions of satisfactory working it would only allow us to reduce s by unity.

89. SMALL EFFECT OF VISCOSITY IN PREVIOUS CALCULATIONS REGARDING *NEWARK*.

It might at first sight appear that what has been said regarding viscosity, to a large extent modifies the earlier conclusions regarding the increase of stress arising from negative values of τ, especially in very stiff gears. But that is true only to a very slight extent. The greatest rate at which energy can be communicated to the actual synchronizing vibration per revolution occurs when $\tau = -4$. Equation (108) shows that if the rate at which energy is absorbed by viscosity is to equal that of its communication by friction for this value of τ, we have

$$.002\, fsa_s' = 4F,$$

or

$$F = \frac{fsa_s'}{2000}.$$

Putting $fa_s' = I_s$, and taking $s = 20$, which is a very high value,

$$\frac{I_s}{F} = 100.$$

Now, from line (rr), *Table X*, we see that for the various values of F and I_s there given, I_s/F ranged from 5.01 to 15.06, the higher value being altogether improbable. Since in the *Newark* I_s had an excessive value, we see that in most cases

where there is sliding contact the action of friction would far outweigh that of viscosity. As remarked earlier, the action of viscosity has been introduced partly to show the extreme limiting conditions of successful practice; but principally for the light thrown on the many analogous cases of structures submitted to synchronous vibrations in which there is no sliding contact and consequently no energy absorbed by friction.

90. SIMILAR GEARS SIMILARLY STRESSED.

The proposition with regard to similar valve motions similarly stressed, given in *Chapter XII*, may be extended to include the action of viscosity. Whether friction is present or not, practically the whole effect of viscosity when the synchronizing vibration is important, is to reduce its energy and amplitude in a definite small proportion.

CHAPTER XIV.

FACTORS OF SAFETY. CONCLUSION.

91. FREQUENT OCCURRENCE OF HIGH INERTIA STRESS.

In the opening paragraph of the present investigation a much broader problem than that which has been discussed is indicated. But the greater part of the investigation applies with but little modification to the determination of the stresses induced in any elastic structure or machine for which the mode of vibration and natural period are calculable; the parts being submitted to known periodic forces, or the movements which the parts of the machinery would have, if composed of perfectly rigid material, being known. The whole investigation has been directed to establishing the very common existence of much higher stresses than are indicated by the usual modes of calculation. That these greatly increased stresses occur in cases of which the *Newark's* L.P. valve gear may be taken as a type will, I think, be admitted to be beyond doubt, as calculation and actual trial entirely agree. But if the opening paragraph is to be fully justified it must be proved that this important increase of stress is comparatively common.

The strongest evidence can be adduced to support this belief.

That short period vibrations occur is a matter of common observation; indeed, this is the characteristic indication of the proximity of running machinery. I do not refer principally to audible vibrations, but to those sensible to the touch. To one standing on a railway bridge, a passing train produces not only a deafening noise, due largely to a highly complex mode of vibration into which the bridge is thrown, but we have also sensible vibrations of longer period, synchronous or nonsynchronous (forced), very markedly present. While, however, it is well to recognize that the effects referred to exist, the evidence of their importance must be much more definite.

In Wöhler's experiments we have bars submitted to repeated strain and stress, as in the case of a live load in a structure; but in the experiments all effects of inertia and synchronous vibration were practically eliminated. The stresses which the experimental bars are able to withstand should, if the above contention is well founded, be much higher than those, calculated by ordinary methods, to which a piece of the same material can be subjected in a machine. And this is the case.

92. WÖHLER'S EXPERIMENTS.

To present this important question clearly, I have extracted from "Engineering," Vol. XI, some of the results of Wöhler's experiments.

In *Table XIV* are given the particulars of the tensile tests of the bars used in the experiments which are noted in *Table XV.* The percentage contraction of area at fracture is added, being

$$100\left(1 - \frac{\text{Breaking stress per square inch of original section.}}{\text{Breaking stress per square inch of fractured section}}\right)$$

This contraction of area is much smaller than is given by the mild steel of to-day, thus showing a harder quality. The lengths of the test pieces are not noted in "Engineering," hence the percentages of elongation are of no value.

Wöhler submitted bars to the various kinds of strain in specially designed machines which automatically repeated the strain

between desired limits of stress. The number of repetitions required to break the bar was noted. Repeating the same experiment with smaller and smaller stresses, served to discover the maximum load, the application of which could be repeated indefinitely without fracturing the bar. The results in *Table XV* are the maximum number of repetitions in each set of experiments, all tests which did not involve more than three million repetitions of the load being rejected. Even where the bar broke, we know, from other series with unbroken bars, that this number would give practically the conditions which could be maintained indefinitely.

The factor in the last column is the ratio of the ultimate tensile stress (original section), *Table XIV*, to the maximum tension in *Table XV*. It is clearly shown how this factor is affected by the range of stress to which the bar is subjected. When the maximum tension is reduced only to about one-half before being repeated the factor is unity or a little over. When the stresses are fully reversed each time the factor is between 2.50 and 3.50. The steel is hard, but with modern mild steel it is improbable we would have a very different result; the lower elastic limit and greater ductility in the modern against the older steel might be expected to counterbalance.

93. ORDINARY FACTORS OF SAFETY AND NECESSARY FACTORS FROM WÖHLER'S EXPERIMENTS COMPARED.

If the calculations are correct and the material sound, factors at all higher than these should be safe.

The ordinary values of factors of safety calculated from the breaking stress, may be roughly stated as—

For a dead load, . . .	4 to 5.
For a live load, . .	8 to 10.

But when the material is doubtful, when possible increase of stress—accidental or otherwise—has to be allowed for, or if increase of rigidity is desired, the factors of safety may rise much beyond these. In special cases they are intentionally lower.

TABLE XIV.—WÖHLER'S EXPERIMENTS. MATERIAL OF TEST BARS.

Line.	Material of which the test bars were made.	Breaking stress. Lbs. per square inch.		Elongation before fracture in percentages of original length.	Percentages contraction of area at fracture.
		Original section.	Fractured section.		
(a)	Wrought iron cut from axle supplied in 1857 by the Phoenix Company,	47,080	66,126	17.8	28.8
(b)		48,150	71,369	21.8	32.5
(c)	Homogeneous iron bars from Pearson, Coleman and Co.,	61,525	91,699	24.6	32.9
(d)		62,060	115,025	16.2	46.0
(e)		65,270	107,000	21.7	39.0
(f)	Cast steel from axle supplied by Krupp in 1862,	109,675	135.997	12.1	19.4
(g)		112,350	140,384	18.6	20.0
(h)		112,350	142,096	11.7	20.9
(k)	Cast steel from axle supplied by the Bochum Company in 1863,	94,160	139,528	18.3	32.5
(l)		95,230	128,721	19.0	26.0
(m)	Cast steel from axle supplied by Vickers, Sons and Co., in 1868,	65,270	115,988	19.5	43.7
(n)		58,850	127,972	15.8	54.0
(o)	Steel plate by Krupp. Strain applied at right angles to direction of rolling,	75,970	140,705	12.2	46.0
(p)		81,320	142,310	10.8	42.9
(q)	Steel plate by Krupp. Strain applied parallel to the direction of rolling,	71,690	147,660	10.2	51.4
(r)		81,320	140,277	9.5	42.0

TABLE XV.—RESULTS OF SOME

Stresses are in

Kind of strain.	Maximum tension.	Minimum tension.	Maximum compression.
Torsion, . . .	40,660	0	. . .
Torsion, . . .	23,540	. . .	23,540
Tension, . . .	34,240	0	. . .
Tension, . . .	47,080	25,680	. . .
Tension, . . .	51,360	0	. . .
Tension, . . .	85,600	37,450	. . .
Tension, . . .	10,700	0	. . .
Transverse strain, .	32,100	0	. . .
Transverse strain, . .	42,800	0	. . .
Transverse strain, .	53,500	0	. . .

OF WÖHLER'S EXPERIMENTS.

pounds per square inch.

Range of stress.	Number of repetitions.	Bar broken or not.	Material. Line, *Table XIV.*	Factor = Breaking stress, *Table XIV* / Max. tension, *Table XV*
40,660	23,850,000	Not.	(f)	2.70 *
			(g)	2.76 *
			(h)	2.76 *
47,080	19,100,000	Not.	(f)	4.65 *
			(g)	4.77 *
			(h)	4.77 *
34,240	10,141,645	Broken.	(a)	1.375
			(b)	1.41
21,400	4,000,000	Not.	(a)	1.00
			(b)	1.02
51,360	13,600,000	Not.	(f)	2.14
			(g)	2.19
			(h)	2.19
48,150	12,000,000	Not.	(f)	1.28
			(g)	1.31
			(h)	1.31
10,700	7,600,000	Not.	Cast iron.	No test of strength.
32,100	48,200,000	Not.	(a)	1.47
			(b)	1.50
42,800	34,500,000	Not.	(c)	1.44
			(d)	1.45
			(e)	1.525
53,500	40,600,000	Not.	(f)	2.05
			(g)	2.10
			(h)	2.10

TABLE XV.—

Kind of strain.	Maximum tension.	Minimum tension.	Maximum compression.
Transverse strain, . .	48,150	0	. . .
Transverse strain, .	53,500	0	. . .
† Transverse strain, .	58,500	0	. . .
Transverse strain applied to rotating bar fixed at one end and loaded at the other. The strain was therefore reversed each half revolution. ‡	17,120	. . .	17,120
	25,680	. . .	25,680
	32,100	. . .	32,100
	32,100	. . .	32,100
	32,100	. . .	32,100
	29,960	. . .	29,960
	27,820	. . .	27,820
	23,540	. . .	23,540
	10,700	. . .	10,700

CONTINUED.

Range of stress	Number of repetitions.	Bar broken or not.	Material. Line, *Table XIV.*	Factor = Breaking stress, *Table XIV* / Max. tension, *Table XV*
48,150	43,000,000	Not.	(k)	1.96
			(l)	1.98
53,500	4,100,000	Not.	(o)	1.42
			(p)	1.52
58,500	3,600,000	Not.	(q)	1.23
			(r)	1.39
34,240	132,250,000	Not.	(a)	2.75
			(b)	2.81
51,360	3,930,150	Broken.	(c)	2.40
			(d)	2.42
			(e)	2.54
64,200	4,163,375	Broken.	(f)	3.42
			(g)	3.50
			(h)	3.50
64,200	45,050,000	Broken.	(f)	3.42
			(g)	3.50
			(h)	3.50
64,200	57,360,000	Not.	(k)	2.93
			(l)	2.97
59,920	3,558,700	Broken.	(k)	3.14
			(l)	3.18
55,640	14,176,171	Not.	(k)	3.38
			(l)	3.42
47,080	8,660,000	Not.	(m)	2.77
			(n)	2.50
21,400	19,327,460	Broken.	Copper.	No test of strength.

TABLE XV—CONTINUED.

FOOT NOTES.

* In these two experiments the stresses are actually shearing stresses; that is, tension accompanied by equal compressions across normal planes. As the compression assists the tension to fracture the material, the ultimate shearing stress is about .7 of the ultimate tension. Hence the factors given, being obtained from tests (f) (g) (h) on ultimate tension, must be corrected by reduction to .7 of the value in the table. This gives

$$2.70 \times .7 = 1.89$$
$$2.76 \times .7 = 1.932$$
$$4.65 \times .7 = 3.255$$
$$4.77 \times .7 = 3.339$$

† There is a misprint in "Engineering," the maximum tension being given as 58,500 pounds and the range of stress as 53,500 pounds. I have given the larger figure, as it is probable that the plate would stand more fatigue parallel to the direction of rolling than at right angles to it, as in the previous experiment.

‡ There were other experiments in this class to show the effect of sudden change of diameter of the shaft, without a fillet. These were decidedly inferior in strength and durability to those given here.

Now, usually the stresses will not be fully reversed in any piece subjected to a live load, the tensions being more intense than the compressions, or *vice versa*. This has been fully pointed out in § 84. In bolts, caps of bearings, and many other parts, the stress can, generally, only come in one direction.* Hence for these cases we would conclude from *Table XV* that the highest necessary factor is about 3, and that frequently a lower one should serve. But there are reasons, apart from any questions of vibration, why these factors should be somewhat increased—the calculation of stress may not be exhaustive; there will be an augmentation of the maximum stress where there is a sudden change of section in any part; in a structure with redundancy of parts it is impossible to anticipate exactly how the stress will be distributed, as, for instance, in a riveted joint the rivets will not all contribute equally to the strength. But even if the calculations are exhaustive and all causes of

* It is shown in § 70 that where extension will upset the running conditions seriously, the maximum allowable stress per square inch on mild steel bolts is from 8 to 12 tons—roughly a factor of 3 from the breaking stress. This is not the condition contemplated in the text, but it leads to the same conclusion.

undue stress are avoided carefully in the design, we know from experience that a factor of safety from a live load of anywhere near 3 would certainly lead to failure in the vast majority of cases. Between 3 and the apparent factor of 8 to 10 there is a great gap which the causes just mentioned are not nearly able to fill; yet, even with these large factors, fractures showing good sections of metal are far from unknown.

The apparent discrepancy has always appealed to me very strongly; as, I have no doubt, it must have done to many others also.

94. REASONS FROM PRECEDING INVESTIGATION FOR HIGH FACTORS OF SAFETY.

But we have seen how, under various conditions, we are liable to a large increase of stress through the induced vibration of natural period. If the friction is larger than the rigid gear inertia in a reciprocating (or rotating) part driven by an elastic gear there is at reversal an important increase of the inertia stress, as shown in *Chapter V.* So great is this that with the rigid gear inertia only one-fourth of the friction the total stress is raised to $2\frac{1}{4}$ times the friction. This result follows however stiff the gear may be. That this increase of stress is real, cannot be doubted; but I do not suppose that anyone watching a valve gear driving a flat slide valve ever saw any visible evidence of it.

Again, Fig. 21 shows, that when the friction is small compared with the inertia, however high the natural period, the stress of the actual synchronizing vibration will (under the most favorable circumstances) rise to nearly 1.4 times that due to all the non-synchronizing terms before τ ceases to be negative; and this with single reversals, so that the effect would, probably, be quite invisible. Before the actual synchronizing vibration ceased to increase τ would have to acquire a definite positive value. The range of conditions which would allow I_s to rise to equality with the inertia stress of all the nonsynchronizing terms before τ became positive, is so wide that the probability of their being fulfilled is great. Hence, even without an important rigid gear synchronizing term, the total inertia stress may be run up.

If the epoch of the actual synchronizing vibration had been near ϵ, Fig. 30, it would have required much more friction than that in the last column of *Table X* to keep I_s so low as 1.2 I_e.

As the value of τ varies so much with the epoch of the actual synchronizing vibration, and depends also on the exact form of the energy transfer curve, there will always be anomalous cases when inertia greatly exceeds friction—some gears may be strained much more heavily than others under, apparently, almost the same conditions. Evidence of this would be exceedingly interesting but would be difficult to obtain definitely.

Again, with regard to visibility of vibration, it is interesting to note that while in the *Newark's* earlier trials it was very evident, as Fig. 30 would lead us to expect since there must be a marked triple reversal for the n_f curves, on the third trial the vibration does not seem to have been remarked. At .65 cut off, it is improbable that I_s was less than 1.2 I_e and 1.51 I_f, and Fig. 30_4 shows that there was still a triple reversal near *f*, Fig. 29. But the actual synchronizing vibration is then little removed from the conditions for single reversals. Thus we have further support for the truth of the theory in the cessation of visibility of the natural period vibrations just when we would expect.

Lastly, when viscosity alone acts in absorbing energy, we have seen in the preceding chapter how smooth the motion must be, unless the gear or structure is exceedingly stiff, if the inertia stress of the actual synchronizing vibration is not to become important. I have known the cylinders of a large inverted marine engine vibrate so much at full power that it was nearly impossible to take indicator diagrams; and it is rare that in such cases there is not a very sensible vibration.

However, when viscosity acts alone, given sufficient stiffness, we have also seen that the increase of stress will be slight, and is it not the case that in such structures as those engine framings made up wholly of struts and ties, which impart great stiffness, the parts may safely be made very light and the stresses high?

It may appear that the necessary slack in the joints of any gear form, in some way, a difficulty in believing that small vibrations of high period exist, as they might not be transmitted

through the gear. But, especially in high speed gear, the brasses require to be set up very closely, in order that the gear may run at all. Slack would produce shock, the energy of which would appear as excessive vibration. Such a gear is, expressively for the present argument, called a "rattletrap." In closely set up bearings, well lubricated, these small vibrations will readily be transmitted; for, if the film of oil in a crank pin serves to separate the surface of the pin from the brass under pressure lasting during half a revolution, it will readily bear the momentary push of a high period vibration without escaping.

If the increase of stress is established, we see that in many successful machines the actual stresses will be nearly, or even more than, double the rigid gear stresses. The necessary factor 3 from Wöhler's experiments will be raised to 5 or 7 for a wholly live load; and the margin between 5 or 7 to a factor of safety of 8 or 10 seems a reasonable allowance for the various causes of local rise of stress, imperfections of workmanship, and so on.

But even with a factor of 10 there is not a great margin for the careless underestimation of stresses ordinarily included. Care should be taken by the adoption of a stiff formation in the design and by the employment of a sufficient factor of safety that ample rigidity is secured. We thus as a rule diminish all these inertia effects and also avoid, in the large majority of cases, the necessity for the more troublesome calculations of elasticity. That such calculations should not always be lightly set aside this whole investigation goes to prove.

Let me enforce the opinions I have put forward by a quotation from a man who is well qualified to speak:

"All who have stood near the center of a 200-foot or even a 100-foot girder, at the time a train has been passing over the same, cannot fail to have noticed that, in addition to a vertical component of motion, a strongly pronounced lateral swinging has taken place. A similar movement is produced by a breeze, which may be considerably increased in a gale, especially perhaps when a train is passing. It was under these conditions that the Tay Bridge disaster took place. The ease with which a transversal swing may be established and then augmented,

even on a 200-foot girder, is recognized by anyone who, standing near the middle of such a structure, has swayed from side to side in a direction transverse to the length of the bridge, timing his movements to those he is creating." *

95. FACTOR OF SAFETY FOR A DEAD LOAD.

The value of the factor of safety for a nearly dead load, above that indicated in *Table XV*, is partly due to the reasons indicated above which go to raise the factor for a live load, such as change of section. There is, besides the necessity to avoid too close proximity to the elastic limit, or serious elastic deformation. But the table shows that a higher actual stress may be employed than for a live load. In other words, the reason for the smaller factor of safety necessary for a dead load lies partly in the physical constitution of the material. This is not what is usually taught or believed, as the following quotation goes to show:

"A *dead load* is a load which is put on by imperceptible degrees, and which remains steady, such as the weight of a structure, or of the fixed framing in a machine. A *live load* is one that is or may be put on suddenly, or accompanied with vibration; like a swift train traveling over a railway bridge; or like most of the forces exerted by and upon the moving pieces in a machine.

"It can be shown that in most cases which occur in practice a live load produces, or is liable to produce, *twice*, or very nearly twice, the effect, in the shape of stress and strain, which an equal dead load would produce. The *mean* intensity of the stress produced by a suddenly applied load is no greater than that produced by the same load acting steadily; but in the case of the suddenly applied load, the stress begins by being insensible, increases to double its mean intensity, and then goes through a series of fluctuations, alternately below and above the mean, accompanied by vibration of the strained body. Hence the

* *Vibrations and Engineering*, by Mr. John Milne, F. R. S., F. G. S., &c. "Engineering," Vol. LXI, page 337. See also, *On the Vibrations Caused by Trains Passing Over Iron Bridges*. By C. A. Pownall, M. Inst., C. E., and John Milne, F. R. S., F. G. S., &c. "Engineering," Vol. LXI, page 111.

ordinary practice is to make the factor of safety for a live load *double* the factor of safety for a dead load."*

The references to vibration as a cause of increase of stress are very interesting, but the passage seems open to much criticism. Equation (22) shows that most of the forces exerted upon the moving parts of a machine, where we have reasonable stiffness, do not exert a much higher maximum stress than an equally intense dead load, unless there is synchronism. Then the limit of increase of stress may be much higher or lower than twice, depending on the conditions of the motion.

What Rankine founds on in the second paragraph is that if a weight is held so as just to touch the scale pan of a frictionless spring balance and, the pointer being at zero, the weight is released, the balance will register, as a maximum, twice the gravity of the weight. This, however, is a special case, and by varying the conditions any desired intensity of stress may be obtained. In any analogous case in practice a careful engineer would calculate the highest stress, not the gravity of the weight alone, and apply his factor of safety to this.

96. CONCLUSION.

It would be absurd to suppose that all the processes of calculation I have given can be frequently repeated in the ordinary course of designing; but they might, at least partly, be resorted to with advantage and with but little labor in cases where the effect of elasticity seems doubtful. Their application would have saved many expensive breakdowns in the past; and if applied to such cases as the *Newark*, where serious trouble has arisen, would gradually accumulate a store of valuable data which could not fail to be useful.

The rigid gear inertia of any motion can be at once obtained, and the smoothness of the motion to some extent judged by the rise of stress over simple harmonic motion. This is a valuable, but not an infallible criterion of the importance of the harmonics, as the acuteness of the maximum depends not only on the amplitude of the various harmonics, but on their epochs also. For

* See Rankine's *Machinery and Millwork*, p. 488, 3d edition.

instance, the same series of harmonics gives the low as well as the high maximum of B_1, Fig. 26, and also all the values intermediate between the maximum. But we may be sure that if the excess is near 86 per cent., as in the *Newark's* full ahead L.P. gear, it would require very great friction to make the gear work if the natural period is not much higher than the 6th. From the *Chicago* we may also safely conclude that for $s = 6$ and the excess over simple harmonic motion, not exceeding 20 per cent., the gear will work with very moderate friction. *Table XIII* almost certainly gives the upper limit of stiffness necessary in the two cases, friction being absent. We have—

For *Newark*, full ahead, or equivalent motion, $j = .305$, $s = 11$ or 12.

For *Chicago*, full ahead, or equivalent motion, $j = .199$, $s = 8$ or 9.

Probably the lower value of s would be satisfactory; and, with very slight friction, at least one period less.

The *Petrel* and *Katahdin* seem to teach that even with an excess of only 30 per cent. it would not be safe to lower s below 6.

The calculation of the natural period for a complex gear involves considerable work; but a rough estimate, taking only the effect of crossbending and torsion, will be easily made, and will indicate if s is dangerously low.

The complete calculation of inertia stress involves the full determination of the rigid gear synchronizing vibration, and this will only be possible in a very limited number of gears; but the energy absorption curve can always be drawn for any value of n_e, and this will show if there are any parts of it with a dangerously low value of τ. A value of s giving such a curve for values of n_e which would raise I_e dangerously, should be avoided if possible. We might be so unfortunate as to have the epoch of the actual synchronizing vibration falling on this part of the curve.

I shall borrow another sentence from Mr. Milne* as a fitting close to this investigation.

"As to the results which may accrue by extending our knowledge of the vibrations of bridges, steamships, buildings and various structures, inasmuch as we are only yet entering a new

* "Engineering," Vol. LXI, p. 399.

field of inquiry, it is impossible to foretell; but from the little we have already gleaned, it is not unlikely that the study will be accompanied with fruitful returns."

Appendix A.

NOTE ON ZEUNER'S DIAGRAM.

The diagram, Fig. 1, is not completed near the center O, in order to call marked attention to the fact that the only way to give a continuous smooth curve through O would be to treat displacements on one side of the arbitrary point O in the valve travel as positive and those on the other as negative, which would make the two parts $abc \ldots$, $a'b'c' \ldots$, fall nearly over one another and confuse the diagram if O is chosen, as Zeuner always does, near the center of the valve motion. Hence, it is convenient to treat in the diagram all displacements, whether on one side of O or the other, as positive.

This change of convention as to the sign of the radius vectors is of no consequence, as the method of estimating stress about to be explained in the test fails at the point O, and it is inadvisable to use it when the radius vector is very short. As the point O is arbitrary the stresses occurring in the parts of the valve movement which would be represented by the parts of the curves omitted may be readily investigated, if thought necessary, by choosing a new point from which to measure displacements. But if O is chosen near the center of motion of the valve, the inertia stress will be far from a maximum in its neighborhood unless the irregularities of the valve motion are very great indeed. The change in the sign of the radius vector is pointed out by Zeuner and is adopted by him simply to keep the diagram clear. See p. 14, footnote, of his work on *Valve Gears;* 3d edition, English translation.

Appendix B.

Effect of various errors on the estimation of the rigid gear inertia stress.

Let us consider the effect on the estimation of the stress of errors in φ and ρ, and also the effect on the final error of the transformations of the valve curve proposed in the text.

If instead of the true value of *sec* φ the measurements give $(1+e)$ *sec* φ, e being the proportionate error, the right-hand side of equation (5) is changed to

$$\frac{r(1+e)^3}{\rho} sec^3\varphi - 2(1+e)^2 sec^2\varphi + 1.$$

As e will always be a small fraction, we may write this,

$$\frac{r}{\rho} sec^3\varphi - 2\,sec^2\varphi + 1 + e\left(\frac{3r}{\rho} sec^3\varphi - 4\,sec^2\varphi\right). \quad . \quad . \quad (1b)$$

$$= \frac{r}{\rho} sec^3\varphi - 2\,sec^2\varphi + 1$$

$$+ e\left\{3\left(\frac{r}{\rho} sec^3\varphi - 2\,sec^2\varphi + 1\right) + 2\,sec^2\varphi - 3\right\}$$

$$= \left\{\frac{r}{\rho} sec^3\varphi - 2\,sec^2\varphi + 1\right\}$$

$$\left\{3 + \frac{r(2\,sec^2\varphi - 3)}{r\left(\frac{r}{\rho} sec^3\varphi - 2\,sec^2\varphi + 1\right)}\right\}. \quad . \quad . \quad . \quad . \quad . \quad (2b)$$

The proportionate error in the deduced stress is thus—equations (2b) and (5)—given by

$$e\left\{3 + \frac{r(2\,sec^2\varphi - 3)}{-\frac{d^2r}{d\theta^2}}\right\}. \quad . \quad . \quad . \quad . \quad . \quad (3b)$$

The smaller φ is the smaller will be the probable value of e. For instance, if φ is actually 8°, the error involved in estimating $\varphi = 0°$—a much larger error than could occur—would only give $e = -.01$, and for $\varphi = 4°$ the error in taking $\varphi = 0°$ would be a quarter of this, and so on. Hence, the advantage of the process

explained in the text, by which φ can always be made nearly zero at the part of the curve where the stress is being measured.

Dealing now with the factor of e, equation (1b) shows at once that the final error in estimation of the stress can only be zero when

$$\frac{3r\ sec^3\varphi}{\rho} - 4\ sec^2\varphi = 0.$$

That is, when

$$\frac{r}{\rho\ cos\ \varphi} = \frac{4}{3}.$$

If we suppose O chosen at the center of the valve travel we may make comparison with the Zeuner diagram. In it we readily see that

$$\frac{r}{\rho\ cos\ \varphi} = 2$$

always; and thus at any point in the exact diagram where the stress is much augmented by the departures from simple harmonic motion, this quantity must be greater than 2. Hence, in all cases of practical importance, an error in estimating $sec\ \varphi$ will always produce an error in the final result.

Expression (3b) shows that when $sec^2\varphi = 1.5$ (φ about 35°) the proportionate error in the final result will be $3e$ wherever O is chosen. For greater values of φ the factor of e will be greater than 3, and for lesser values less, but not evanescent except for very low stress. Thus, when $\varphi = 0$ the proportionate error in equation (26) becomes

$$e\left(3 - \frac{1}{r/\rho - 1}\right),$$

when O is near the center of the valve travel, r/ρ in all important cases is, as we have just seen, greater than 2, and the proportionate error will still be greater than $2e$.

We conclude from the above that large values of φ should be avoided, as they conduce both to a large probable value of e and to a large factor.

Let us now examine the effect on the proportionate error of shifting the arbitrary zero point O.

$$Tan\,\varphi = \frac{1}{r}\frac{dr}{d\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4b)$$

Hence, as $dr/d\theta$ is not affected by a shift of the zero point, $tan\,\varphi$ is inversely proportional to r; and the farther from the point under examination we choose O, the nearer zero φ will be, and the smaller the probable value of e.

The variable part of the factor of e is

$$+\frac{r(2\,sec^2\varphi - 3)}{-\frac{d^2r}{d\theta^2}}.$$

As Fig. (1) is drawn $d^2r/d\theta^2$ is negative and r is always assumed positive, hence the factor will be increased by an increase in the algebraic value of $r(2\,sec^2\varphi - 3)$, and decreased by a decrease.

If $$y = r(2\,sec^2\varphi - 3),$$

we have $$\frac{dy}{dr} = 2\,sec^2\varphi - 3 + 4r\,sin\,\varphi\,sec^3\varphi\frac{d\varphi}{dr}.$$

From equation (4b),

$$(1 + tan^2\varphi)\frac{d\varphi}{dr} = -\frac{1}{r^2}\frac{dr}{d\theta} = -\frac{tan\,\varphi}{r},$$

since $dr/d\theta$ is constant;

and $$\frac{d\varphi}{dr} = -\frac{sin\,\varphi\,cos\,\varphi}{r}.$$

Thus $$\frac{dy}{dr} = 2\,sec^2\varphi - 3 - 4\,sin^2\varphi\,sec^2\varphi = 1 - 2\,sec^2\varphi.$$

This quantity is always negative, and therefore the factor of e will always be diminished by removing the zero point O from the part of the curve under examination if $d^2r/d\theta^2$ is negative. In other words, errors arising from a definite proportionate error in the estimation of φ will be decreased at any point a', Fig. 1, by shifting out from O all the points in the curve $a'b'c'$ . . . in the neighborhood of a', by a constant amount.

In Fig. 2, shifts inward greater than the radius vector are used,

but as they are made when φ is zero, e will be zero, and thus they are not open to errors of estimation of φ.

Consider now the effect of an error in measuring ρ. If e_1 is the proportionate error, the calculated stress will be

$$\frac{r}{\rho(1+e_1)} sec^3\varphi - 2\,sec^2\varphi + 1;$$

or approximately

$$\frac{r}{\rho} sec^3\varphi - 2\,sec^2\varphi + 1 - e_1 \frac{r\,sec^3\varphi}{\rho}.$$

It will be difficult to measure ρ within a small fraction of its whole amount if it is either very small or very great, and therefore it will depend on the particular case what the probable value of e_1 is and whether a transformation in the diagram will increase or diminish it. As stated in the text and shown at the end of this note, ρ is usually diminished by changing the curve to make φ zero, and it is increased by shifting the curve out from O. Unless for unusually large valve diagrams decrease of e_1 would probably follow from making these changes simultaneously if the curve was carried out till ρ was at least 2 to 3 inches in length.

From equation (5) the factor of e_1

$$\frac{r\,sec^3\varphi}{\rho} = -\frac{1}{r}\frac{d^2r}{d\theta^2} + 2\,sec^2\varphi - 1.$$

For $d^2r/d\theta^2$ negative and, of course, constant, the first term is positive and is diminished by shifting out the curve. $2\,sec^2\varphi - 1$ is also positive and is diminished both by shifting out the curve and reducing φ to zero.

To sum up—The probable accuracy of the process is increased both by changing the curve so that φ will become small and by adding a constant length to all the radius vectors near the point under examination. Against this must be set any inaccuracy of the processes by which the changes are effected, but these are of such a nature that they may be performed with great exactness. There is also a limit to the amount by which the radius vectors may be profitably increased, as the radial shifting of the points

separates them; if too far apart, the exact form of the curve will become more doubtful. Trial, however, seems to bear out the advantage of using these shifts in many cases; but it is, of course, a matter which must be left to individual judgment.

The truth of the two statements in the text regarding the changes of radius of curvature when φ is reduced to zero and when then the reference point O is shifted, is almost evident. It is perhaps well, however, to set down more exactly the nature of the changes and the limitation of the first statement.

Equation (5) may be put in the form—

$$\frac{1}{\rho} = \frac{\cos\varphi}{r}\left\{2 - \left(\frac{1}{r}\frac{d^2r}{d\theta^2} + 1\right)\cos^2\varphi\right\}.$$

We need only consider the case of $d^2r/d\theta^2$ negative and r positive, as in Fig. 1.

If r is unchanged and the curve turned around so as to change φ, we have,

$$r\frac{d}{d\varphi}\frac{1}{\rho} = -2\sin\varphi + 3\left(\frac{1}{r}\frac{d^2r}{d\theta^2} + 1\right)\sin\varphi\cos^2\varphi.$$

Equating this to zero gives

$$\sin\varphi = 0 \text{ or } 3\left(\frac{1}{r}\frac{d^2r}{d\theta^2} + 1\right)\cos^2\varphi = 2. \quad . \quad . \quad . \quad (5b)$$

1st. $$-\frac{1}{r}\frac{d^2r}{d\theta^2} > \tfrac{1}{3}.$$

It will be found that $\varphi = 0$ makes

$$\frac{d^2}{d\varphi^2}\frac{1}{\rho} \text{ negative.}$$

Therefore ρ is then a minimum.

The value of φ for ρ a maximum is given by

$$3\left(\frac{1}{r}\frac{d^2r}{d\theta^2} + 1\right)\cos^2\varphi = 2.$$

As this gives $\cos^2\varphi > 1$ no maximum if ρ is reached for a finite change of $dr/d\theta$.

2d. $$-\frac{1}{r}\frac{d^2r}{d\theta^2} < \tfrac{1}{3}.$$

The above maximum and minimum are now reversed. The second equation (5b) now gives, for the position in which ρ is a minimum, a real value of φ which recedes from zero as

$$-\frac{1}{r}\frac{d^2r}{d\theta^2}$$

is diminished.

In Zeuner's diagram

$$r = a_1 \cos(\theta - \varepsilon_1)$$

and

$$\frac{d^2r}{d\theta^2} = -a_1 \cos(\theta - \varepsilon_1) = -r,$$

or

$$-\frac{1}{r}\frac{d^2r}{d\theta^2} = 1. \quad \ldots\ldots\ldots \quad (6b)$$

Here, if φ were changed from a comparatively large angle to zero, ρ would become a good deal smaller and the probable value of ϵ_1 might be much increased.

Again, equation (6b) shows that for the Zeuner circle diagram the point O would have to be shifted from the point (r_1, θ_1) of the curve till all the radius vectors had been increased by a constant amount greater than $2r_1$ before we reached a set of curves for which

$$-\frac{1}{r}\frac{d^2r}{d\theta^2} \text{ is less than } \tfrac{1}{3}.$$

Till then, $\varphi = 0$ gives ρ a minimum at this point. In the exact diagram, at points of specially high stress, $-\frac{1}{r}\frac{d^2r}{d\theta^2}$ will be greater than 1, if O is near the center of the valve travel, and $\varphi = 0$ would give ρ a minimum over a correspondingly greater change of the radius vectors than in Zeuner's diagram, which is a larger shift than would usually be employed.

For comparatively large shifts, the above indicates that the valve of ρ, and consequently the probable value of ϵ_1 ceases to be affected much by changing φ to zero.

If $\varphi = 0$, we get,

$$\frac{1}{\rho} = \frac{1}{r}\left(1 - \frac{1}{r}\frac{d^2r}{d\theta^2}\right),$$

$d^2r/d\theta^2$ being negative, ρ evidently increases with r.

The questions treated in this appendix may seem to have more

space devoted to them than their importance demands, but the estimation of the stresses is a process so very delicate that the effect on the accuracy of the result arising from any of the processes involved deserves careful attention.

Appendix C.

HARMONIC ANALYSIS OF THE MOTION OF THE POINT *A*, FIGS. 4 TO 7.

As the analysis of the motion of the point *A*, Figs. 4 to 7, throws much light on the action of valve gears, the method by which equations (12) to (15) were obtained is given here.

As explained in the text, the variable height of *A* is given by

$$y = k + a\cos\theta. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1c)$$

The harmonic analysis of the horizontal movement is required.

$$r = \sqrt{(R^2 - y^2)} = R\sqrt{\left(1 - \frac{y^2}{R^2}\right)};$$

that is,

$$\frac{r}{R} = \sqrt{\left\{1 - \frac{k^2}{R^2}\left(1 + \frac{a}{k}\cos\theta\right)^2\right\}}. \quad . \quad . \quad . \quad (2c)$$

As y/R is always less than unity, we may expand equation (2c) by the Binominal Theorem.

The expansion may be written,

$$\frac{r}{R} = 1 - N_2\left(1 + \frac{a}{k}\cos\theta\right)^2 - N_4\left(1 + \frac{a}{k}\cos\theta\right)^4 - N_6\left(1 + \frac{a}{k}\cos\theta\right)^6 - \&c., \quad . \quad (3c)$$

where $N_2 = \frac{1}{2}\frac{k^2}{R^2}$, $N_4 = \frac{1 \cdot 3}{2^3 \underline{|2}}\frac{k^4}{R^4}$, $N_6 = \frac{1 \cdot 3 \cdot 5}{2^3 \underline{|3}}\frac{k^6}{R^6}$, &c.

For successive calculation these coefficients are most conveniently expressed thus—

$$N_2 = \frac{1}{2}\frac{k^2}{R^2},\ N_4 = \frac{3}{2}N_2^2,\ N_6 = \frac{5}{3}N_2N_4,\ N_8 = \frac{7}{4}N_2N_6, \ . \ . \ .$$

$$. \ . \ . \ N_n = \frac{2(n-1)}{n}N_2N_{n-2} \ . \ . \ . \ . \ . \ . \ . \ . \quad (4c)$$

The expansion of equation (3c) in terms of powers of $\cos\theta$ is given in the following table.

TABLE Ic.

	-1	$-\frac{a}{k}\cos\theta$	$-\left(\frac{a}{k}\cos\theta\right)^2$	$-\left(\frac{a}{k}\cos\theta\right)^3$	$-\left(\frac{a}{k}\cos\theta\right)^4$	$-\left(\frac{a}{k}\cos\theta\right)^5$	$-\left(\frac{a}{k}\cos\theta\right)^6$	—&c.
1	-1	. .	. .	. .	. .	. .	. .	. .
N_2	1	2	1	. .	. .	. .	. .	. .
N_4	1	4	6	4	1	. .	. .	. .
N_6	1	6	15	20	15	6	1	. .
N_8	1	8	28	56	70	56	28	&c.
N_{10}	1	10	45	120	210	252	210	&c.
&c.	&c.	&c.	&c.	&c.	&c.	&c.	&c.	&c.
$\frac{r}{R}=$	P_0	$-P_1\left(\frac{a}{k}\cos\theta\right)$	$-P_2\left(\frac{a}{k}\cos\theta\right)^2$	$-P_3\left(\frac{a}{k}\cos\theta\right)^3$	$-P_4\left(\frac{a}{k}\cos\theta\right)^4$	$-P_5\left(\frac{a}{k}\cos\theta\right)^5$	$-P_6\left(\frac{a}{k}\cos\theta\right)^6$	&c.

Each of the numbers in the horizontal rows is to be multiplied by the N coefficient in the left-hand column and in the same row; and all the products in the vertical columns are to be added to find the P coefficient at the foot of the same column. Thus,

$$P_4 = N_4 + 15\,N_6 + 70\,N_8 + 210\,N_{10} + \&c.$$

As

$$P_0 = 1 - N_2 - N_4 - N_6 - \&c.,$$

we see from equation (3c) that it is equal to r/R when $a = 0$. Therefore, from equation (2c),

$$P_0 = \sqrt{\left(1 - \frac{k^2}{R^2}\right)},$$

and can be estimated without making the summation indicated in *Table Ic.*

We must now apply the well-known formulas—

$$2\cos^2\theta = 1 + \cos 2\theta,$$
$$2^2\cos^3\theta = 3\cos\theta + \cos 3\theta,$$
$$\&c. \qquad \&c. \qquad \&c.,$$

which for convenience we may arrange in a table thus—

TABLE IIc.

	1	$\cos\theta$	$\cos 2\theta$	$\cos 3\theta$	$\cos 4\theta$	$\cos 5\theta$	$\cos 6\theta$	&c.
$2\cos^2\theta =$	1	. .	1	. .	. .	. .	. .	. .
$2^2\cos^3\theta =$	. .	3	. .	1	. .	. .	. .	. .
$2^3\cos^4\theta =$	3	. .	4	. .	1	. .	. .	. .
$2^4\cos^5\theta =$	. .	10	. .	5	. .	1	. .	. .
$2^5\cos^6\theta =$	10	. .	15	. .	6	. .	1	. .
&c.	. .	&c.	. .	&c.	. .	&c.	. .	&c.
&c.	&c.	. .	&c.	. .	&c.	. .	&c.	. .

Arranged in this way the law of the numerical coefficients becomes evident and the table can readily be extended indefinitely.

We can now arrange the final result in a table constructed like *Table Ic.*

TABLE IIIc.

	1	$\cos\theta$	$\cos 2\theta$	$\cos 3\theta$	$\cos 4\theta$	$\cos 5\theta$	$\cos 6\theta$	&c.
P_0	1	..	..	..	..	..	..	..
$-\frac{P_1}{1}\left(\frac{a}{k}\right)$	..	1	..	..	..	..	..	..
$-\frac{P_2}{2}\left(\frac{a}{k}\right)^2$	1	..	1	..	..	..	..	..
$-\frac{P_3}{2^2}\left(\frac{a}{k}\right)^3$	..	3	..	1	..	..	..	..
$-\frac{P_4}{2^3}\left(\frac{a}{k}\right)^4$	3	..	4	..	1	..	..	..
$-\frac{P_5}{2^4}\left(\frac{a}{k}\right)^5$	..	10	..	5	..	1	..	..
$-\frac{P_6}{2^5}\left(\frac{a}{k}\right)^6$	10	..	15	..	6	..	1	..
&c.	..	&c.	..	&c.	..	&c.	..	&c.
&c.	&c.	..	&c.	..	&c.	..	&c.	..
$\frac{r}{R}=$	Q_0	$-Q_1\cos\theta$	$-Q_2\cos 2\theta$	$-Q_3\cos 3\theta$	$-Q_4\cos 4\theta$	$-Q_5\cos 5\theta$	$-Q_6\cos 6\theta$	$-$ &c.

The operations indicated by this table are readily intelligible after what has been said explanatory of *Table Ic.* Thus,

$$Q_2 = \frac{P_2}{2}\left(\frac{a}{k}\right)^2 + \frac{4\,P_4}{2^3}\left(\frac{a}{k}\right)^4 + \frac{15\,P_6}{2^5}\left(\frac{a}{k}\right)^6 + \&c.$$

This process fails when $k = 0$. We then have

$$\frac{r}{R} = \sqrt{\left\{1 - \frac{a^2}{R^2}\cos^2\theta\right\}}$$
$$= 1 - N_2\cos^2\theta - N_4\cos^4\theta - N_6\cos^6\theta - \&c.,$$

the only change in equations (4c) being that now

$$N_2 = \frac{1}{2}\frac{a^2}{R^2}.$$

The rest of the numerical process can readily be arranged in tables similar to those just given.

Examination of the evolution of the Q coefficients shown by *Tables Ic* and *IIIc* at once makes evident the reason of the great response of the amplitude of the high period movements to any change in the angle *COX* of Figs. 4 to 7, or in the angular swing *EOF* of the links *OA*. Separating out the commencement of the series for Q_1, Q_2 and Q_3, we have—

$$Q_1 = \frac{a}{R}\left\{\frac{k}{R} + \frac{3}{2}\left(\frac{k}{R}\right)^3 + \frac{15}{8}\left(\frac{k}{R}\right)^5 + \ldots\right\}$$
$$+ \left(\frac{a}{R}\right)^3\left\{\frac{9}{8}\left(\frac{k}{R}\right) + \frac{75}{16}\left(\frac{k}{R}\right)^3 + \frac{735}{64}\left(\frac{k}{R}\right)^5 + \ldots\right\}$$
$$+ \&c.$$

$$Q_2 = \left(\frac{a}{R}\right)^2\left\{\frac{1}{4} + \frac{9}{8}\left(\frac{k}{R}\right)^2 + \frac{75}{32}\left(\frac{k}{R}\right)^4 + \ldots\right\}$$
$$+ \left(\frac{a}{R}\right)^4\left\{\frac{3}{16} + \frac{75}{32}\left(\frac{k}{R}\right)^2 + \frac{1225}{128}\left(\frac{k}{R}\right)^4 + \ldots\right\}$$
$$+ \&c.$$

$$Q_3 = \left(\frac{a}{R}\right)^3\left\{\frac{3}{8}\frac{k}{R} + \frac{25}{16}\left(\frac{k}{R}\right)^3 + \frac{245}{64}\left(\frac{k}{R}\right)^5 + \ldots\right\}$$
$$+ \left(\frac{a}{R}\right)^5\left\{\frac{75}{128}\frac{k}{R} + \frac{1225}{256}\left(\frac{k}{R}\right)^3 + \frac{19845}{1024}\left(\frac{k}{R}\right)^5 + \ldots\right\}$$
$$+ \&c.$$

Thus, the amplitude of high period motions depend only on high powers of the ratio a/R; and principally on high powers of the ratio k/R, especially if the ratios a/R and k/R are comparatively large. The series then become very slowly convergent. This is well shown by the fact that to obtain the coefficients for the first line of *Table VI* ($k/R = .8$, $a/R = .12$), it was necessary to carry the calculation as far as N_{60} and P_{11}: even then several remainders had to be estimated, and there is a slight doubt, in some cases, whether it would not be more accurate to replace the last decimal given by the figure above or below it.

Appendix D.

MOTION OF A MASS SUSPENDED BY A SPRING, THE UPPER END OF WHICH HAS ANY ARBITRARY VERTICAL PERIODIC MOTION.—(1) NO RESISTANCE TO MOTION. (2) FRICTION PROPORTIONAL TO VELOCITY.

To prevent inconvenient interruption of the text, I give in this Appendix the deduction of some results quoted there, relating to the effect of elasticity of the gear.

A mass M is suspended by a spring of known strength. The upper end is moved vertically in a known manner, and the consequent movement of M is required.

(1) *No Resistance to Motion.*

If suspension of M were by a rigid connection, it would have the same movement as the upper end of the spring.

r = the displacement of M, in feet, at time t, measured from any chosen position in its line of motion, when the spring is replaced by a rigid connection.

r_1 = the actual displacement of M, in feet, at time t.

f = the strength of the spring, in pounds weight, for a known value of g, the intensity of gravity. Thus the force in absolute units required to displace the mass M by a distance δr, when the upper end of the spring is held fixed, is $fg\delta r$.

If r and r_1 are estimated as positive when the displacement is upward, $r_1 - r$ positive will give a downward force on the

mass M. Hence the acceleration will be negative. Thus, supposing no friction, the equation of motion is

$$\left.\begin{aligned} M\frac{d^2r_1}{dt^2} + fg(r_1 - r) &= 0, \\ \text{or} \qquad \frac{d^2r_1}{dt^2} + \omega_s^2(r_1 - r) &= 0, \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (1d)$$

where $\omega_s^2 = fg/M$, a positive quantity.

Equation (1) we may write, for brevity,

$$r = k + \Sigma_1^{\infty} a_n \cos(n\omega t - \varepsilon_n), \quad . \quad . \quad . \quad . \quad (2d)$$

n being put equal to each positive whole number in succession.

From the physical problem involved we may readily perceive the nature of the solution of equation (1d).

When the engine is standing with $r = 0$, if the valve—mass M—is forcibly displaced in its line of motion and released, it will vibrate indefinitely with its natural period, since there is supposed to be no friction or other cause to absorb the energy of vibration. Equation (1d) then reduces to

$$\frac{d^2r_1}{dt^2} + \omega_s^2 r_1 = 0,$$

of which the known solution is

$$r_1 = A\cos(\omega_s t - \lambda), \quad . \quad . \quad . \quad . \quad . \quad . \quad (3d)$$

where A and λ are two independent arbitrary constants. That is, the motion imposed on M may have any amplitude and may pass through its central position at any time. (Practically, the amplitude would be confined within that determined by the elastic limit of the gear or spring if a permanent change in the initial conditions were not to result, but the solution does not express this, as no such limitation has been supposed in forming equation (1d)).

We may note here that this solution (3d) may also be put in the form

$$r_1 = A'\cos\omega_s t + B'\sin\omega_s t, \quad . \quad . \quad . \quad . \quad (3_1d)$$

where A' and B' are independent arbitrary constants. For we have

$$A' = A \cos \lambda, \; B' = A \sin \lambda;$$

that is, $A'^2 + B'^2 = A^2$ and $\tan \lambda = B'/A'$.

Thus we may choose A and λ so as to give A' and B' any values we please.

Equation (3d) will form part of the solution of equation (1d), since the motion expressed by equation (2d) does not prevent any disturbance from affecting the motion of M, nor does it alter the rigidity, f, of the gear or spring.

There is left the part of the movement of M which is induced by the working of the gear or movement of the upper part of the spring. When the gear is running steadily it must repeat the same movements each revolution of the engine. The motion must therefore consist of a series of simple harmonic motions with periods equal to the period of the engine or submultiples of this period; that is, it must be similar in form to equation (2d).

Thus we have

$$r_1 = k' + \Sigma_1^\infty a_n' \cos (n\omega t - \varepsilon_n') + A \cos (\omega_s t - \lambda), \quad . \; . \; . \; (4d)$$

and $$\frac{d^2 r_1}{dt^2} = -\Sigma_1^\infty a_n' n^2 \omega^2 \cos (n\omega t - \varepsilon_n') - A\omega_s^2 \cos (\omega_s^2 t - \lambda). \quad (5d)$$

Substituting for r, r_1, and $d^2 r_1/dt^2$—given by equations (2d), (4d), and (5d)—in equation (1d), we get,

$$\begin{aligned} &-\Sigma_1^\infty a_n' n^2 \omega^2 \cos (n\omega t - \varepsilon_n') - A\omega_s^2 \cos (\omega_s t - \lambda) \\ &\quad + \omega_s^2 \{k' + \Sigma_1^\infty a_n' \cos (n\omega t - \varepsilon_n') + A \cos (\omega_s t - \lambda) \\ &\qquad - k - \Sigma_1^\infty a_n \cos (n\omega t - \varepsilon_n)\} = 0. \quad . \; . \quad (6d) \end{aligned}$$

If equation (4d) is to be a solution of equation (1d), this expression must be an identity. $A\omega_s^2 \cos (\omega_s t - \lambda)$ vanishes identically. The constant quantities must cancel one another; hence

$$k' = k.$$

We are left with a series of periodic functions, and in order

that equation (6d) may be fulfilled identically, each set of periodic functions of the same period must vanish identically. Hence

$$a_n'(\omega_s^2 - n^2\omega^2)\cos(n\omega t - \varepsilon_n') = a_n\omega_s^2\cos(n\omega t - \varepsilon_n). \quad (6_1\text{d})$$

The two sides of this equation pass through their zero value at the same if

$$\varepsilon_n' = \varepsilon_n,$$

and will have the same maximum value if

$$a_n' = \frac{a_n\omega_s^2}{\omega_s^2 - n^2\omega^2}. \quad (7\text{d})$$

Let us now express ω_s and $n\omega$ in terms of the corresponding periodic times T and T_n. When M is vibrating in its natural periodic time the motion is expressed by $A\cos(\omega_s t - \lambda)$. This goes through a complete cycle in time T; that is, the angle $\omega_s t - \lambda$ increases by 2π in that time. Thus,

$$\omega_s T = 2\pi, \text{ or } \sqrt{\frac{fg}{M}} = \omega_s = \frac{2\pi}{T},$$

which agrees with equation (16).

Similarly, $$n\omega T_n = 2\pi, \text{ or } n\omega = \frac{2\pi}{T_n},$$

which agrees with equation (21).

Substituting these values of T and T_n in equation (7d), we get

$$a_n' = \frac{a_n\left(\frac{2\pi}{T}\right)^2}{\left(\frac{2\pi}{T}\right)^2 - \left(\frac{2\pi}{T_n}\right)^2} = \frac{a_n}{1 - T^2/T_n^2}.$$

Equation (4d) thus reduces to

$$r_1 = k + \Sigma_1^{\infty}\frac{a_n}{1 - T^2/T_n^2}\cos(n\omega t - \varepsilon_n) + A\cos(\omega_s t - \lambda). \quad (8\text{d})$$

This agrees with equation (22) and completes the solution of equation (1d) except for an exact coincidence between the natural

period T and that of any one of the periodic disturbances produced by the engine. If

$$T_n = T,$$

the coefficient

$$\frac{a_n}{1 - T^2/T_n^2}$$

becomes infinite, and the term in equation (8d),

$$\frac{a_n}{1 - T^2/T_n^2} cos\,(n\omega t - \varepsilon_n),$$

ceases to be part of the solution of equation (1d); as, supposing such a coincidence, the corresponding periodic motion could not become indefinitely great all at once. But there would be a gradual communication of energy and gradual growth of this disturbance.

It is instructive to examine the nature of the solution for this case, and the result will be found of the very highest importance, not only in the solution of the simple problem where friction is supposed absent, but in the actual physical problem with friction taken into account.

As the solution, equation (8d), holds good except for one period we only require the solution of equation (1d) when

$$r = a_s cos\,(\omega_s t - \varepsilon_s), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9d)$$

s being the suffix indicating all quantities belonging to the synchronizing vibration. Thus, a_n, $n\omega$, and ε_n are replaced by a_s, ω_s, and ε_s.

Suppose that $\omega_s t - \varepsilon_s = \omega_s t'$, and instead of making the periods coincide exactly make

$$r = a_s cos\,(\omega_s - \gamma_1)\, t'. \quad . \quad . \quad . \quad . \quad . \quad . \quad (10d)$$

By taking γ_1 small enough we may make T_n as nearly as we please equal to T, and thus watch the effect of gradual approach to synchronism.

From equations (7d) and (8d) the solution of equation (1d), for the term (10d), is

$$r_1 = \frac{a_s\omega_s^2}{\omega_s^2 - (\omega_s - \eta)^2}\cos(\omega_s - \eta)\,t' + A\cos(\omega_s t' - \lambda)^*$$
$$= \left\{ A\sin\lambda + \frac{a_s\omega_s^2 \sin\eta t'}{\eta(2\omega_s - \eta)} \right\} \sin\omega_s t'$$
$$+ \left\{ A\cos\lambda + \frac{a_s\omega_s^2 \cos\eta t'}{\eta(2\omega_s - \eta)} \right\} \cos\omega_s t'. \quad . \quad . \quad (11d)$$

Let us now choose η so small that, for any period of time over which we desire to follow the motion, $\eta t'$ remains a small angle. We have then, sensibly,

$$\cos\eta t' = 1, \ \sin\eta t' = \eta t'.$$

If also we write $2\omega_s$ for $2\omega_s - \eta$, since the ratio $\eta/2\omega_s$ is so minute, equation (11d) reduces to

$$r_1 = \left\{ A\sin\lambda + \frac{a_s\omega_s t'}{2} \right\} \sin\omega_s t'$$
$$+ \left\{ A\cos\lambda + \frac{a_s\omega_s}{2\eta} \right\} \cos\omega_s t'. \quad . \quad . \quad (12d)$$

η only appears now in the term $a_s\omega_s/2\eta$, but as η is diminished towards zero to establish synchronism, this quantity becomes indefinitely large. We must examine the case carefully.

Make the substitution

$$A\sin\lambda = C,$$

where C is an arbitrary constant.

Then $$A\cos\lambda = C\cot\lambda,$$

and equation (12d) becomes

$$r_1 = \left(C + \frac{a_s\omega_s t'}{2}\right)\sin\omega_s t' + \left(C\cot\lambda + \frac{a_s\omega_s}{2\eta}\right)\cos\omega_s t'. \quad . \quad (13d)$$

Hence for any value of C required in the coefficient of $\sin\omega_s t'$, we may so choose λ that however large $a_s\omega_s/2\eta$ is

$$C\cot\lambda + \frac{a_s\omega_s}{2\eta} = A',$$

where A' is a perfectly arbitrary constant.

* The change from t to t' makes a change in λ; but as λ is arbitrary the change does not require to be noted.

Equation (13d) becomes,

$$r_1 = \left(C + \frac{a_s\omega_s t'}{2}\right) sin\ \omega_s t' + A' cos\ \omega_s t';$$

or, replacing t,

$$r_1 = \left(C - \frac{a_s\varepsilon_s}{2} + \frac{a_s\omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) + A' cos\,(\omega_s t - \varepsilon_s)$$
$$= \left(B_0 + \frac{a_s\omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) + A' cos\,(\omega_s t - \varepsilon_s), \quad . \quad . \quad (14d)$$

where $B_0 = C - \frac{a_s\varepsilon_s}{2}$ is the value of the coefficint of $sin\,(\omega_s t - \varepsilon_s)$ when $t = 0$.

For the complete value of r_1 we must add the right-hand sides of equations (8d) and (14d), the term for $n = s$ being omitted in the summation term of the former. The sum of the two terms

$$A\,cos\,(\omega_s t - \lambda) + A'\,cos\,(\omega_s t - \varepsilon_s)$$

may, by altering the values of the arbitrary constants A and λ, be given by the single term $A\,cos\,(\omega_s t - \lambda)$. Or, as shown in equation (25_2), § 15,

$$A\,cos\,(\omega_s t - \lambda) = B_0\,sin\,(\omega_s t - \varepsilon_s) + A_0\,cos\,(\omega_s t - \varepsilon_s).$$

Hence we may write the complete solution of equation (1d), when there is synchronism

$$r_1 = k + \Sigma_1^{\infty} \frac{a_n}{1 - T^2/T_n^2} cos\,(n\omega t - \varepsilon_n)$$
$$+ \left(B_0 + \frac{a_s\omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) + A\,cos\,(\omega_s t - \lambda), \quad . \quad . \quad (15d)$$

or

$$r_1 = k + \Sigma_1^{\infty} \frac{a_n}{1 - T^2/T_n^2} cos\,(n\omega t - \varepsilon_n)$$
$$+ \left(B_0 + \frac{a_s\omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) + A\,cos\,(\omega_s t - \varepsilon_s), \quad . \quad . \quad (16d)$$

B_0 and A having somewhat different values in the two equations, and $n = s$ being omitted in the summation term.

Equation (16d) is the more instructive form. As noted in § 15, it shows that, whatever the amplitue and epoch of the actual synchronizing vibration, the moment synchronism is established, the resolved part of it which lags one quarter period behind the rigid gear synchronizing term $a_s \cos(\omega_s t - \varepsilon_s)$ begins to increase in amplitude at a uniform rate; while the part which has the same epoch is unaffected. That is, the term $a_s \cos(\omega_s t - \varepsilon_s)$ communicates energy only to that part of the actual synchronizing vibration which follows behind it by one quarter period, and communicates no energy to the portion agreeing with it in phase.

The amplitude of the term $(B_0 + a_s\omega_s t/2)\sin(\omega_s t - \varepsilon_s)$ at time t_1 is $B_0 + a_s\omega_s t_1/2$. After one cycle it becomes

$$B_0 + \frac{a_s}{2}(\omega_s t_1 + 2\pi).$$

Hence the mean force exerted in making this increase is that due to the mean value of the above amplitudes, or

$$f\left\{B_0 + \frac{a_s}{2}(\omega_s t_1 + \pi)\right\}.$$

The total increase of amplitude during the cycle is $a_s\pi$. And thus the energy communicated during the cycle beginning at t_1 is

$$f a_s \pi \left\{B_0 + \frac{a_s}{2}(\omega_s t_1 + \pi)\right\}. \quad . \quad . \quad . \quad . \quad (17d)$$

which is expression (25).

As this expression is of fundamental importance it is well to get a perfectly clear view of the exact process by which the energy is transferred. As the gear if rigid would give the vibration of natural period a displacement of $a_s \cos(\omega_s t - \varepsilon_s)$, while it has actually a displacement of

$$\left(B_0 + \frac{a_s\omega_s t}{2}\right)\sin(\omega_s t - \varepsilon_s) + A\cos(\omega_s t - \varepsilon_s),$$

the elastic force exerted by the valve, due to this difference, is always

$$f\left\{ a_s \cos(\omega_s t - \varepsilon_s) - \left(B_0 + \frac{a_s \omega_s t}{2}\right) \sin(\omega_s t - \varepsilon_s) - A \cos(\omega_s t - \varepsilon_s) \right\} \cdot \quad (18d)$$

The motion of the valve under this force in time dt, is

$$\frac{d}{dt}\left\{ \left(B_0 + \frac{a_s \omega_s t}{2}\right) \sin(\omega_s t - \varepsilon_s) + A \cos(\omega_s t - \varepsilon_s) \right\} dt. \quad (19d)$$

The energy transferred in time dt is the product of expressions (18d) and (19d). Integrating the expressions thus obtained between the limits of time for one cycle

$$\omega_s t_1 - \varepsilon_s = 2n\pi \text{ and } \omega_s t_1 - \varepsilon_s + 2\pi = 2(n+1)\pi$$

we again obtain expression (17d). I do not set down the integration as it presents no difficulty, but would occupy considerable space.

If the limits of integration are different from the above the arbitrary constant A will not disappear; showing that while the total energy transferred between the beginning of one cycle and the next is not affected by the presence of the arbitrary vibration $A \cos(\omega_s t - \varepsilon_s)$, the exact manner of transfer of energy is affected by it.

There is one other point which might cause difficulty. If a mass M was revolving at the end of a constantly increasing arm of length $B_0 + a_s \omega_s t/2$, its radial motion would constitute a portion of its energy. But as the augmentation of amplitude is constant the quantity of this energy is also constant and hence does not need to be considered in the transfer of energy. In the actual reciprocating motion of the valve we have, at each point, the portion of this velocity due to the increase of arm resolved in the direction of motion of the valve; and as this is the same at the beginning and end of each cycle, wherever that cycle commences, it does not affect the transfer per cycle.

Let us now take the case in *Chapter VIII*, where J and J' are small and the mean amplitude a_s' is treated as if it were constant during the cycle. Since α is the angle by which the actual synchronizing vibration departs from normal epoch, the resolved

portion which has normal epoch is $a_s' \cos \alpha$. Substituting this for the mean amplitude

$$B_0 + \frac{a_s}{2}(\omega_s t_1 + \pi)$$

in expression (17d) gives the tranfer of energy per cycle

$$f a_s a_s' \pi \cos \alpha, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20d)$$

which is expression (88).

Expression (20d) also follows from an integration like that indicated in connection with expression (17d), which in this case is

$$f \int \{a_s \cos(\omega_s t - \varepsilon_s) - a_s' \sin(\omega_s t - \varepsilon_s \pm \alpha)\} \frac{d}{dt} a_s' \sin(\omega_s t - \varepsilon_s \pm \alpha)\, dt, \quad . \quad (21d)$$

the limits being the beginning and end of a cycle starting at any time t.

In the actual problem f is not alway constant, but is a function of θ and therefore of t. In the *Newark* the variation of f, as shown in § 64, is nearly 34 per cent. above and below its mean value.

To consider f as a harmonic function of θ in treating equation (1d) leads to a great complication. Each period in the rigid gear movement now produces a whole series of vibrations at different periods in the actual motion of M, instead of only one. To determine the amplitudes and epochs of the actual motion the equations of condition can readily be obtained; but, instead of the single equation (6_1d), there is an infinite series in each of which all the required amplitudes and epochs appear. So far as I have been able to discover, the only method of solving equation (1d) with f variable is to solve these equations of condition. I think there can be no doubt that this can be effected by approximations in any particular case, but one is unwilling to undertake it without some definite and important object in view, as the calculations would be very elaborate.

I think it may be safely concluded that at particular speeds of engines with $F = 0$, there would, during each revolution, be a

transfer of energy from the gear to the mass M, causing a continually increasing vibration synchronizing with one of the rigid gear vibrations. This vibration would be accompanied in its increase by a series of others less intense, producing altogether a vibration of continuously varying amplitude and period, the latter of which follows more or less closely the variations of the reciprocal of the square root of f—see equations (16) and (17). On account of the interaction of all of the periods, when F is finite the resulting largely magnified vibration—which corresponds to the synchronizing vibrations for f constant—could not be simply represented by a series of small circles and crosses as in Figs. 20 and 29, spaced irregularly; or by replacing the normal vector Oa, Fig. 16, by one revolving with a variable velocity. Thus the determination of the rates of energy transfer would be most tedious.

But as, in the simpler problems, departures from normal epoch by diminishing $\cos \alpha$ involve a diminished rate of energy transfer from the gear, variations of f will almost certainly have the same effect. Even in the case of the *Newark*, however, where f varies so much, this effect must be small—entirely negligible compared with the uncertainty regarding the value of F. Thus, taking the mean value of δr, as in § 64, and treating f as constant must lead to almost the same results as the far more complex solution of the exact problem.

(2) *Friction Proportional to Velocity.*

Let V in pounds be the friction when the velocity is one foot per second. A positive velocity dr_1/dt would then induce a retarding force

$$Vg\frac{dr_1}{dt},$$

and this must be added to the force due to the elasticity of the gear in equation (1d). Hence the equation of motion now is,

$$\left.\begin{aligned} M\frac{d^2r_1}{dt^2} + Vg\frac{dr_1}{dt} + fg(r_1 - r) &= 0, \\ \text{or} \qquad \frac{d^2r_1}{dt^2} + \mu\frac{dr_1}{dt} + \omega_s^2(r_1 - r) &= 0, \end{aligned}\right\} \quad . \quad . \quad (22d)$$

where $$\mu = \frac{Vg}{M}.$$

That equations (104) and (105) give the solution of this equation may readily be verified by trial.

Appendix E.

NATURAL PERIOD OF U. S. S. *NEWARK'S* L.P. VALVES.

As the actual case of the *Newark's* L.P. valve gear with two valves, each having its own elasticity, is much more complex than the simple case taken in the text, § 64, it is necessary to investigate the natural period more closely, to insure that no great error has been made by adopting the simpler procedure. It is also of importance to show how to treat the actual case, as the simplification adopted in the text would not always be admissible.

M_1 and M_2 are the respective masses of the two valves V_1 and V_2, Figs. 27 and 28, with their spindles and part of the valve gears, as estimated in § 64.

f_1 and f_2 are the elastic resistances, in pounds, of the two valves. That is, a force $f_1 \delta r$ pounds applied to valve V_1 along its axis, will displace it by a distance δr, valve V_2 being left free and there being no friction. A similar meaning is to be given to f_2.

As before, r is the displacement which either valve would have at time t if the gear were rigid. For simplicity I have supposed r the same for both valves—as it actually is in the *Newark*—though forming the equation for different masses M_1 and M_2. There would be no difficulty in solving the equation with r supposed a different complex harmonic function for each valve, and the reader can readily make the required change. As we are only investigating the natural periods, and not requiring the complete solution of the differential equations of motion, there would be no advantage in the more complex procedure.

Suppose a force applied to V_1 alone moves it by a distance r_1'. As the whole gear is strained the valve V_2 will also move. Let the consequent movement of V_2 be $\gamma_1 r_1'$.

A force being applied to V_2 alone, r_2' and $\gamma_2 r_2'$ have similar meanings.

We have then—

Total displacement of valve V_1 at time $t = r_1 = r + r_1' + \gamma_2 r_2'$.
" " " " V_2 " " $t = r_2 = r + r_2' + \gamma_1 r_1'$. } . (1e)

As the part of the displacement of V_1 due to force applied *to it* is r_1', and as r_2' is the similar displacement of valve V_2, the equations of motion are,

$$\left.\begin{aligned} &\text{For valve } V_1;\ M_1 \frac{d^2 r_1}{dt^2} + f_1 g r_1' = 0, \text{ or } -\frac{1}{\chi_1^2}\frac{d^2 r_1}{dt^2} = r_1'. \\ &\text{For valve } V_2;\ M_2 \frac{d^2 r_2}{dt^2} + f_2 g r_2' = 0, \text{ or } -\frac{1}{\chi_2^2}\frac{d^2 r_2}{dt^2} = r_2'. \end{aligned}\right\} \quad . \quad (2e)$$

Where $\chi_1^2 = \dfrac{f_1 g}{M_1}$ and $\chi_2^2 = \dfrac{f_2 g}{M_2}$.

From equations (1e) and (2e) we then have,

$$\left.\begin{aligned} r_1 &= r - \frac{1}{\chi_1^2}\frac{d^2 r_1}{dt^2} - \frac{\gamma_2}{\chi_2^2}\frac{d^2 r_2}{dt^2} \\ r_2 &= r - \frac{1}{\chi_2^2}\frac{d^2 r_2}{dt^2} - \frac{\gamma_1}{\chi_1^2}\frac{d^2 r_1}{dt^2} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (3e)$$

Rewriting equations (3e),

$$\frac{1}{\chi_1^2}\frac{d^2 r_1}{dt^2} + \frac{\gamma_2}{\chi_2^2}\frac{d^2 r_2}{dt^2} + r_1 - r = 0. \quad . \quad . \quad . \quad . \quad (4e)$$

$$\frac{\gamma_1}{\chi_1^2}\frac{d^2 r_1}{dt^2} + \frac{1}{\chi_2^2}\frac{d^2 r_2}{dt^2} + r_2 - r = 0. \quad . \quad . \quad . \quad . \quad (5e)$$

Differentiating these twice with respect to t, we get,

$$\frac{1}{\chi_1^2}\frac{d^4 r_1}{dt^4} + \frac{\gamma_2}{\chi_2^2}\frac{d^4 r_2}{dt^4} + \frac{d^2 r_1}{dt^2} - \frac{d^2 r}{dt^2} = 0. \quad . \quad . \quad . \quad (6e)$$

$$\frac{\gamma_1}{\chi_1^2}\frac{d^4 r_1}{dt^4} + \frac{1}{\chi_2^2}\frac{d^4 r_2}{dt^4} + \frac{d^2 r_2}{dt^2} - \frac{d^2 r}{dt^2} = 0. \quad . \quad . \quad . \quad (7e)$$

Multiply equation (6e) by χ_1^2,
" " (7e) " $\gamma_2 \chi_1^2$ and subtract,
" " (4e) " $\chi_1^2 \chi_2^2$ and add.

The resulting equation is,

$$(1-\gamma_1\gamma_2)\frac{d^4r_1}{dt^4}+(\chi_1^2+\chi_2^2)\frac{d^2r_1}{dt^2}-\chi_1^2(1-\gamma_2)\frac{d^2r}{dt^2}+\chi_1^2\chi_2^2(r_1-r)=0. \quad . \quad (8e)$$

We have thus eliminated r_2.

As in *Appendix D*, to find the natural periods we may suppose the engine standing with $r=0$, one or both of the valves forcibly displaced in their line of motion, and released equation (8e) then reduces to

$$(1-\gamma_1\gamma_2)\frac{d^4r_1}{dt^4}+(\chi_1^2+\chi_2^2)\frac{d^2r_1}{dt^2}+\chi_1^2\chi_2^2r_1=0. \quad . \quad (9e)$$

This equation will obviously be satisfied by a solution of the form

$$r_1=A\cos(\omega_s t-\lambda). \quad . \quad . \quad . \quad . \quad . \quad (10e)$$

Substituting this in equation (9e) we find

$$(1-\gamma_1\gamma_2)\omega_s^4-(\chi_1^2+\chi_2^2)\omega_s^2+\chi_1^2\chi_2^2=0. \quad . \quad . \quad (11e)$$

Solving this for ω_s^2 we get

$$2(1-\gamma_1\gamma_2)\omega_s^2=\chi_1^2+\chi_2^2\pm\sqrt{\{(\chi_1^2+\chi_2^2)^2-4\chi_1^2\chi_2^2(1-\gamma_1\gamma_2)\}}. \quad . \quad (12e)$$

There are thus two natural periods: we need not distinguish between the positive and negative values of ω_s, got by extracting the square root. From symmetry we see that by eliminating r_1 instead of r_2 we would have arrived at exactly the same equation (12e). Hence the pairs of natural periods of the two valves are the same, as we might expect.

If, as proposed above, we forcibly displace one or both valves and release them, we would set up vibrations of both natural periods. Hence it would not be so easy in this, as in the simple case of one valve, to find the natural period experimentally. The relative displacements which must be given simultaneously to the two valves at release, in order to have only one or other of the natural periods present, can be quite easily calculated; but it does not seem to be of sufficient practical importance to justify troubling the reader with it.

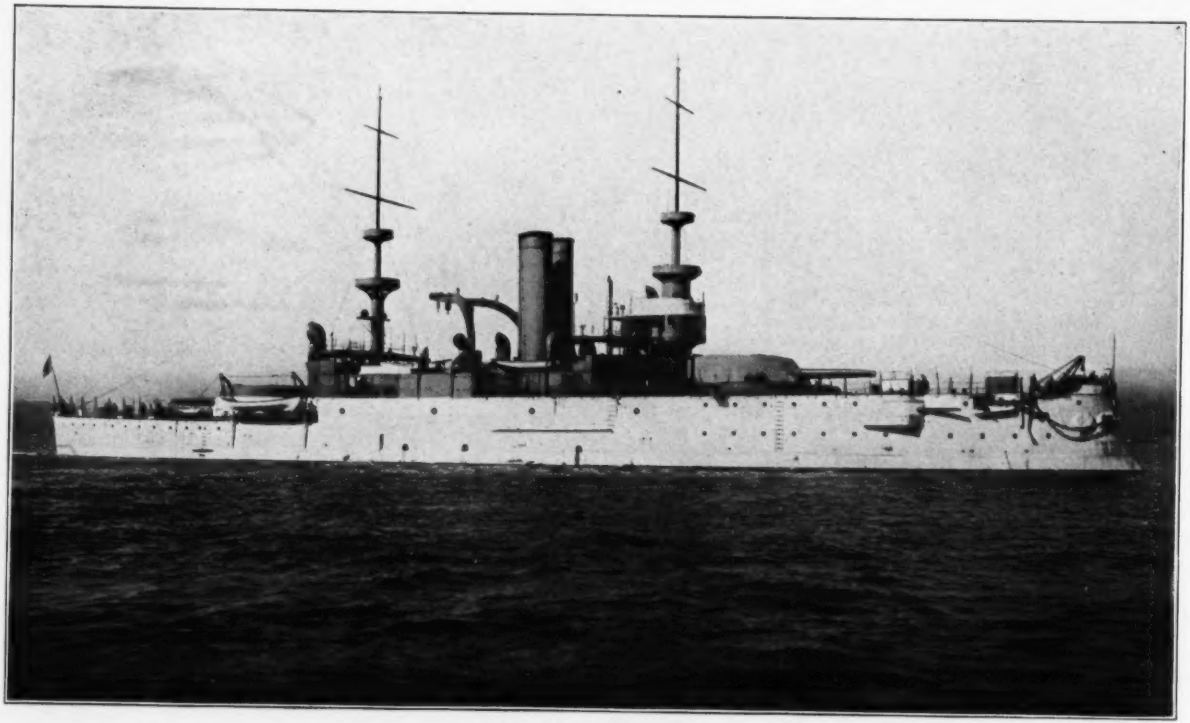
Let us now calculate actual values by use of *Table VIII.*

Line.	Configuration of gear.	Most elastic.		Least elastic.	
(a)	Valve,	V_1	V_2	V_1	V_2
(b)	$M_1 = M_2 = \frac{1}{2}M$ in § 64,	1180	1180	1180	1180
(c)	δr, in feet, *Table VIII*, $\frac{10^{-6}}{12} \times$	36.336	37.696	18.290	19.650
(d)	$f = \frac{1}{\delta r}$, in pounds, $10^6 \times$	$f_1 = .33025$	$f_2 = .3183$	$f_1 = .6561$	$f_2 = .6107$
(e)	$\chi_1^2 = \frac{f_1 g}{M_1}$, $\chi_2^2 = \frac{f_2 g}{M_2}$,	$\chi_1^2 = 9012$	$\chi_2^2 = 8686$	$\chi_1^2 = 17{,}904$	$\chi_2^2 = 16{,}665$
(f)	Consequent yield of other valve in feet, *Table VIII*, $\frac{10^{-6}}{12}$	34.755	34.317	16.709	16.271
(g)	γ_1 and γ_2 = Line (f) ÷ line (c),	$\gamma_1 = .9565$	$\gamma_2 = .9104$	$\gamma_1 = .9136$	$\gamma_2 = .8280$
(h)	$\gamma_1 \gamma_2$,	.8708		.7565	
(k)	$1 - \gamma_1 \gamma_2$,	.1292		.2435	
(l)	$\chi_1^2 + \chi_2^2$,	17698		34569	
(m)	$(\chi_1^2 + \chi_2^2)^2$,	$10^8 \times 3.13219$		$10^8 \times 11.9502$	
(n)	$4\chi_1^2\chi_2^2(1 - \gamma_1\gamma_2)$,	$10^7 \times 4.0453$		$10^8 \times 2.9060$	
(o)	$\sqrt{\{(\chi_1^2 + \chi_2^2)^2 - 4\chi_1^2\chi_2^2(1 - \gamma_1\gamma_2)\}}$,	16516		30074	
(p)	ω_s^2, from equation (12e),	132408 or 4574		132744 or 9230.5	
(q)	ω_s, radians per second,	364 or 67.6		364 or 96.1	
(r)	Number of vibrations per minute = $60\omega_s/2\pi$,	3476 or 646		3476 or 918	
(s)	Number of natural period of vibration when the engine is running 127 revolutions per minute = Line (r) ÷ 127,	27.4 or 5.09		27.4 or 7.23	

This entirely confirms the calculation of natural period in § 64. Accompanying the natural period used in the text we have a second of very much shorter period.

If the reader solves the equations of motion completely he will find that if there is synchronism with either of the natural periods, the same characteristics as distinguish equation (16d) present themselves. That is, the actual synchronizing vibration grows in amplitude uniformly with the time, and its normal vector lags one quarter period behind that of the rigid gear vibration.

Possibly the short period vibration is of little practical importance in this case. But we must remember that if, when it is evanescent, its epoch is such that τ is negative and, also, that energy is transmitted from the gear, the stress from it is bound to grow till one or both of these conditions change. It would be futile attempting to investigate the stress from the 27th period vibration, as we have done from the 6th. For it cannot be hoped that the calculations of f_1, f_2, γ_1 and γ_2 are so accurate that we are justified in saying more than that the high period vibration is in the neighborhood of the 27th. Also, very slight errors in the measurements of E° and F° will greatly affect the result by altering the curves of energy transfer and change of epoch from friction, and shifting the normal epoch. As the maximum tension (or compression) on the valve spindle due to this vibration occurs about every 13° of crank angle we may be sure that if of sensible magnitude it goes to increase the maximum stress calculated in the text. We must also carefully bear in mind that the stress due to it would be at least equal to the total stress of all the non-synchronizing terms combined before it would produce a triple reversal. As this is probably the point at which a vibration becomes evident we must not conclude that it is unimportant because we cannot easily perceive it.



U. S. S. "WISCONSIN."

CONTRACT TRIAL OF THE UNITED STATES BATTLESHIP *WISCONSIN.*

By Lieutenant J. K. Robison, U. S. Navy.

The *Wisconsin* is a twin-screw, armored, seagoing battleship. She was constructed by the Union Iron Works of San Francisco, Cal., under a contract dated September 19, 1896, and at a cost, exclusive of armor and armament, of $2,674,950. Penalties were provided in the contract for any delay in completion beyond three years; for failure to maintain a speed of sixteen knots an hour during a continuous four-hour trial at sea, with an air pressure in the firerooms not exceeding 1 inch of water; also for excess in weight of machinery above 1,130 tons.

The hull and machinery specifications and general plans were supplied by the Navy Department, but the details were developed by the contractors. It is noteworthy that the battleships of the *Wisconsin* class are practically identical with those of the *Kearsarge* class as regards hull and machinery; the chief differences between the two types of battleships lying in the battery and in the distribution of the armor. For convenience, general dimensions of the ship and of the machinery are given, together with plans showing the arrangement of the ship.

GENERAL DIMENSIONS.

Length between perpendiculars, feet and inches	368- 0
Projection forward of forward perpendicular, feet and inches	5-10½
Length over all, feet and inches	373-10½
Beam, extreme, feet and inches	72- 2½
molded, feet and inches	72- 0
Mean draught, feet and inches	23- 6
Displacement to L.W.L., 23 feet 6 inches, tons	11,565
Tons per inch at L.W.L., 23 feet 6 inches, tons	47.8
Depth, base line to top of main deck beams at side, feet and inches	34- 7⅜
Point of ram, below 23 feet 6 inches W.L., feet and inches	8- 9

HEIGHTS ABOVE LOAD-WATER LINE, 23 FEET 6 INCHES.

Fore and main masts, truck lights, feet and inches	120– 3¼
floor, upper top, feet and inches	72– 2⅜
lower top, feet and inches	56– 1⅜
Search-light platform (aft), feet and inches	46– 0⅜
Top of chart house, feet and inches	44– 1⅜
Forward bridge, feet and inches	37– 1⅜
After bridge, feet and inches	34– 7⅜
Top of bridge deck at side, feet and inches	25– 4⅜
upper deck planking at stem, feet and inches	19– 9⅜
main deck planking at stern, feet and inches	13– 4⅜
flat armor deck, feet and inches	3– 7⅜
forward turret, feet and inches	31– 4¾
after turret, feet and inches	23–10¾

DOCKING KEELS.

Total length, feet and inches	196– 0
Center line of ship to outside of keel, feet and inches	20– 3
Forward end, forward of center of forward turret, feet and inches	11– 0
Aft end, abaft center of after turret, feet and inches	8– 0

BILGE KEELS.

Forward keel, length, feet and inches	84– 0
extends between frames	14 and 35
After keel, length, feet and inches	78– 6
extends between frames	59 and 79

BATTERY.

2 13-inch B.L.R. guns in forward turret.
2 13-inch B.L.R. guns in after turret.
Angle of train, 135°, both sides of fore-and-aft line.
8 6-inch B.L.R. guns on main deck inside casemate.
2 6-inch B.L.R. guns on main deck, forward (frame 14).
Angle of train, 93° forward, 45° aft of beam.
4 6-inch B.L.R. guns on upper deck, amidships.
Angle of train of forward pair, 90° forward, 45° aft of beam.
Angle of train of aft pair, 45° forward, 90° aft of beam.
4 6-pounder semi-automatic guns, upper bridge, forward.
4 6-pounder semi-automatic guns, upper bridge, aft.
2 6-pounder semi-automatic guns, bridge deck, amidships.
2 6-pounder R.F. guns, main deck, forward.
4 6-pounder R.F. guns, berth deck, aft.
4 1-pounder automatic guns, military tops.

2 1-pounder R.F. guns on berthing forward bridge deck.
2 3-inch field guns on berthing forward bridge deck.
4 Colt's automatic guns on berthing aft bridge deck.
4 above-water, broadside torpedo tubes.
Angle of train: forward tubes, 30° forward, 26° aft of beam.
after tubes, 30° forward, 30° aft of beam.

AMMUNITION STOWAGE.

Kind of ammunition.	No. in box.	Allowance.	Possible stowage.
13 inch A. P. shell	...	96	96
13-inch S. A. P. shell	...	72	72
13-inch common shell	...	72	72
13-inch powder tanks	...	480	541
6-inch A. P. shell	...	840	1,019
6-inch common shell*	...	1,680	2,191
6-inch shrapnel	...	280	371
6-inch cartridges	1	2,800	2,946
6-pounder cartridges	11	730	1,093
6 pounder saluting powder	11	27	27
6-pounder drill	10	8	8
1-pounder cartridges	60	50	50
1-pounder drill	10	3	3
3-inch field-gun cartridges	17	18	18
6-millimeter colt	1,000	40	40
30-caliber rifle ball	1,000	128	128
30-caliber rifle blank	1,000	10	10
30-caliber rifle dummy	1,000	3	3
45-caliber rifle ball	1,000	10	10
38-caliber revolver ball	3,000	12	12
22-caliber rifle ball	10,000	10	10
Saluting powder	1	17	17
Torpedo heads	1	8	8

* 520 6-inch common shell are stowed in ready-service racks on main and upper decks.

WATER-TIGHT COMPARTMENTS.

Location.	Divisions.				Total.
	A.	B.	C.	D.	
Double bottom	3	8	3	3	17
Hold	23	24	10	22	79
Platform deck	29	...	...	28	57
Splinter deck	11	23	10	13	57
Protective deck	21	...	...	18	39
Berth deck	12	18	7	10	47
Main deck	2	1	...	...	3
Total	101	74	30	94	299

WATER-TIGHT DOORS.

Style.	Hold.	Platform.	Splinter.	Berth.	Main.	Total.
Hinged	37	48	44	47	4	180
Vertical sliding	12	...	...	20	..	32
Total	49	48	44	67	4	212

COAL-BUNKER CAPACITIES.

HOLD.

	Starboard.		Port.	
	Cubic feet.	*Tons.*	*Cubic feet.*	*Tons.*
Bunker No. 1	3,971	92.4	3,971	92.4
Bunker No. 2	3,536	82.3	3,536	82.3
Bunker No. 3	3,731	86.8	3,731	86.8
Bunker No. 4	3,704	86.2	3,704	86.2
Bunker No. 5	3,036	70.7	3,157	73.4
Bunker No. 6	1,388	32.3	1,388	32.3
Bunker No. 7	1,105	25.7	1,105	25.7
Totals	20,471	476.4	20,592	479.1

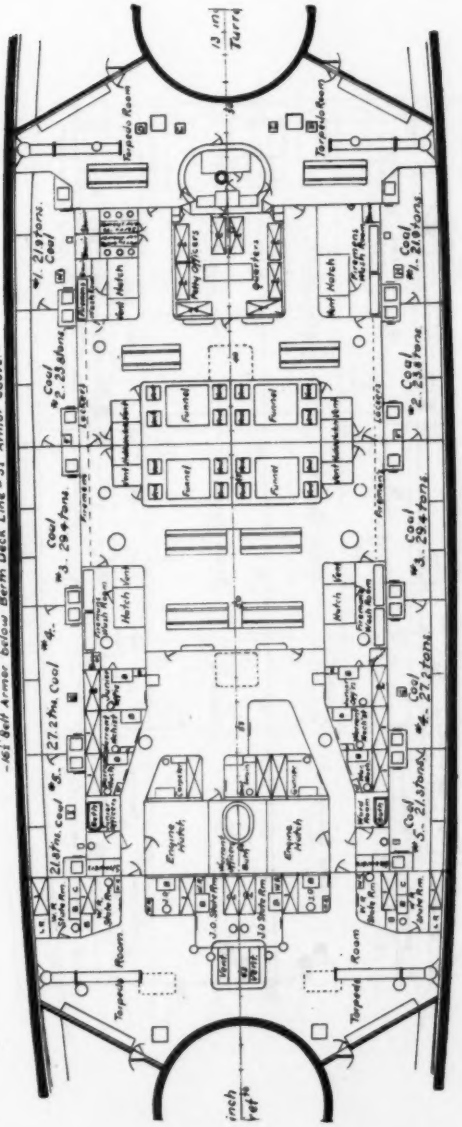
SPLINTER DECK.

Bunker No. 1	653	15.2	406	9.4
Bunker No. 2	772	17.9	772	17.9
Bunker No. 3	689	16.0	562	13.1
Bunker No. 4	582	13.5	220	5.1
Totals	2,696	62.6	1,960	45.5

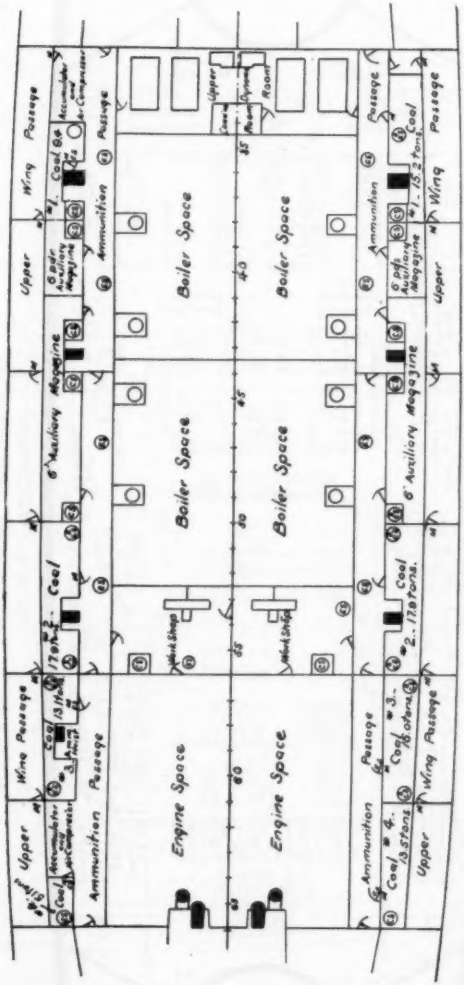
BERTH DECK.

Bunker No. 1	941	21.9	941	21.9
Bunker No. 2	1,024	23.8	1,024	23.8
Bunker No. 3	1,262	29.4	1,262	29.4
Bunker No. 4	1,170	27.2	1,170	27.2
Bunker No. 5	914	21.3	914	21.3
Totals	5,311	123.6	5,311	123.6

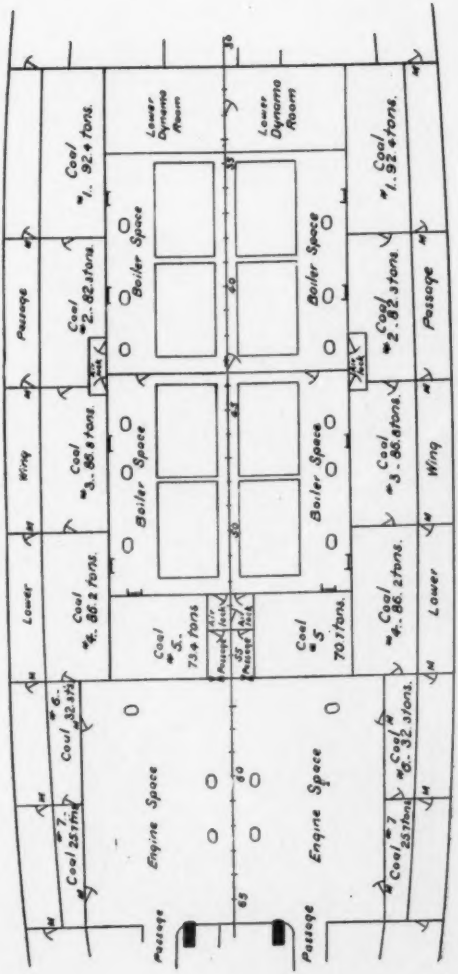
	Cubic feet.	*Tons.*
Hold bunkers	41,063	955.5
Splinter-deck bunkers	4,656	108.1
Berth-deck bunkers	10,622	247.2
Grand total	56,341	1,310.8



U. S. S. "WISCONSIN."
COAL BUNKERS.—BERTH DECK.



U. S. S. "WISCONSIN."
COAL BUNKERS.—SPLINTER DECK.



U. S. S. "WISCONSIN."
COAL BUNKERS.—HOLD.

DOUBLE-BOTTOM COMPARTMENTS.—CAPACITIES.

Compartment.	Location.	Gallons.	Tons, S. W.	Tons, F. W.
A—97	Frames 22 to 25.	13,683.9	52.3	50.8
A—98	Frames 25 to 28.	14,147.7	54	52.5
A—99	Frames 28 to 31.	14,529.6	55.5	53.9
B—92	Frames 31 to 35.	24,666.1	94.2	91.6
B—93	Frames 35 to 38.	18,655.8	71.6	69.6
B—94	Frames 38 to 41.	18,849.6	72	70.0
B—95	Frames 41 to 44.	18,867.6	72.1	70.1
B—96	Frames 44 to 47.	18,939.4	72.3	70.3
B—97*	Frames 47 to 50.	18,216.0	69.6	67.7
B—98*	Frames 50 to 53.	18,158.4	69.4	67.4
B—99	Frames 53 to 56.	18,140.5	69.3	67.4
C—97	Frames 56 to 59.	15,824.7	60.4	58.8
C—98	Frames 59 to 62.	16,327.3	62.4	60.6
C—99	Frames 62 to 66.	23,675.7	90.4	87.9
D—97	Frames 66 to 69.	14,804.4	56.5	54.9
D—98	Frames 69 to 72.	13,553.8	51.8	50.3
D—99	Frames 72 to 75.	12,817.7	48.9	47.6

*R. F. T.

CAPACITY OF TRIMMING TANKS.

	Gallons.	Tons, S.W.	Tons, F.W.
Forward tank, No. 1, comp't A— 1, hold...	3,600	13.75	13.36
Forward tank, No. 2, comp't A— 2, hold...	15,318	58.51	56.88
Forward tank, No. 5, comp't A—24, plat....	3,901	14.90	14.49
Forward tank, No. 6, comp't A—25, plat....	14,507	55.41	53.87
After tank, No. 3, comp't D—21, hold.....	21,541	82.28	80.00
After tank, No. 4, comp't D—22, hold.....	6,941	26.51	25.78
Total.....	65,808	251.36	244.38

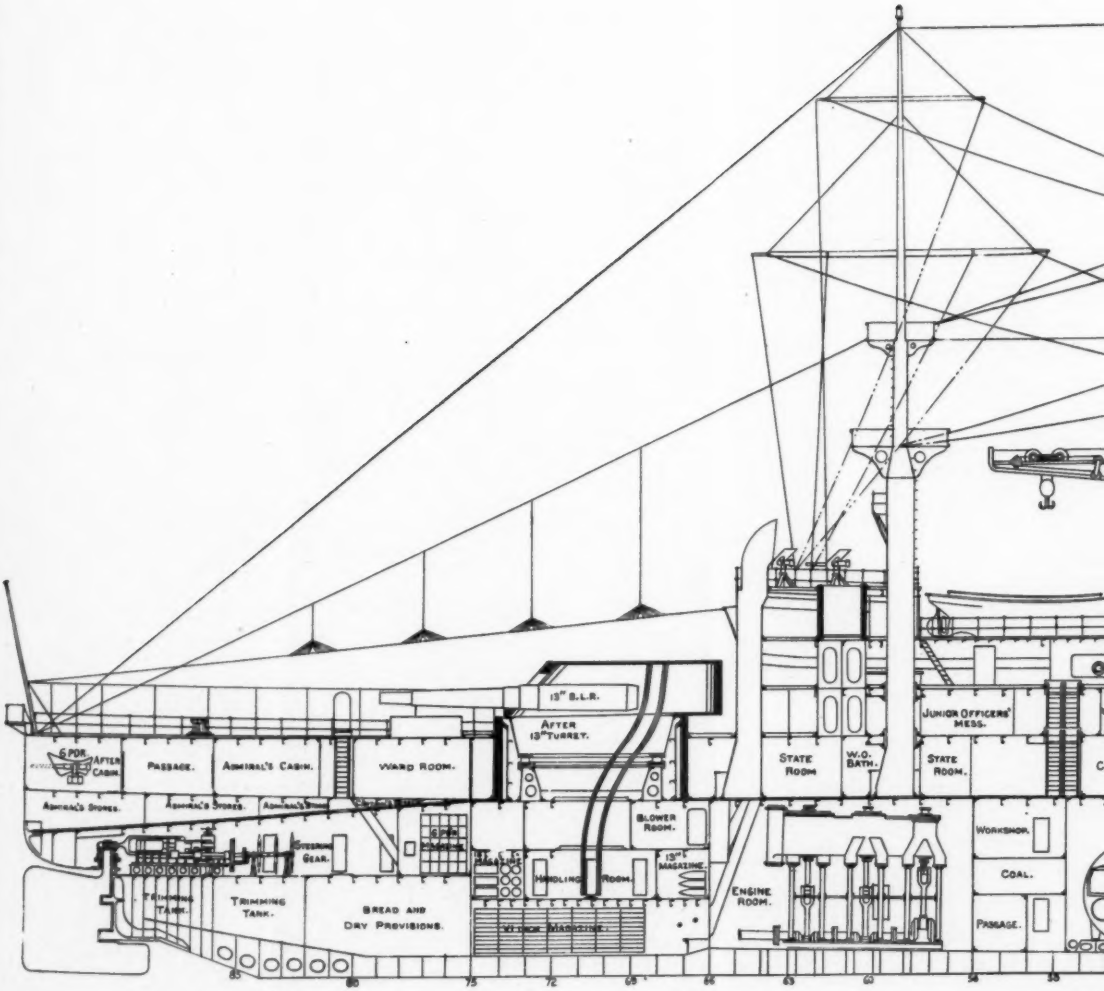
CAPACITY OF FRESH-WATER TANKS.

Compartments A—7 and 8, forward hold, between bulkheads 17 and 22, starboard and port:

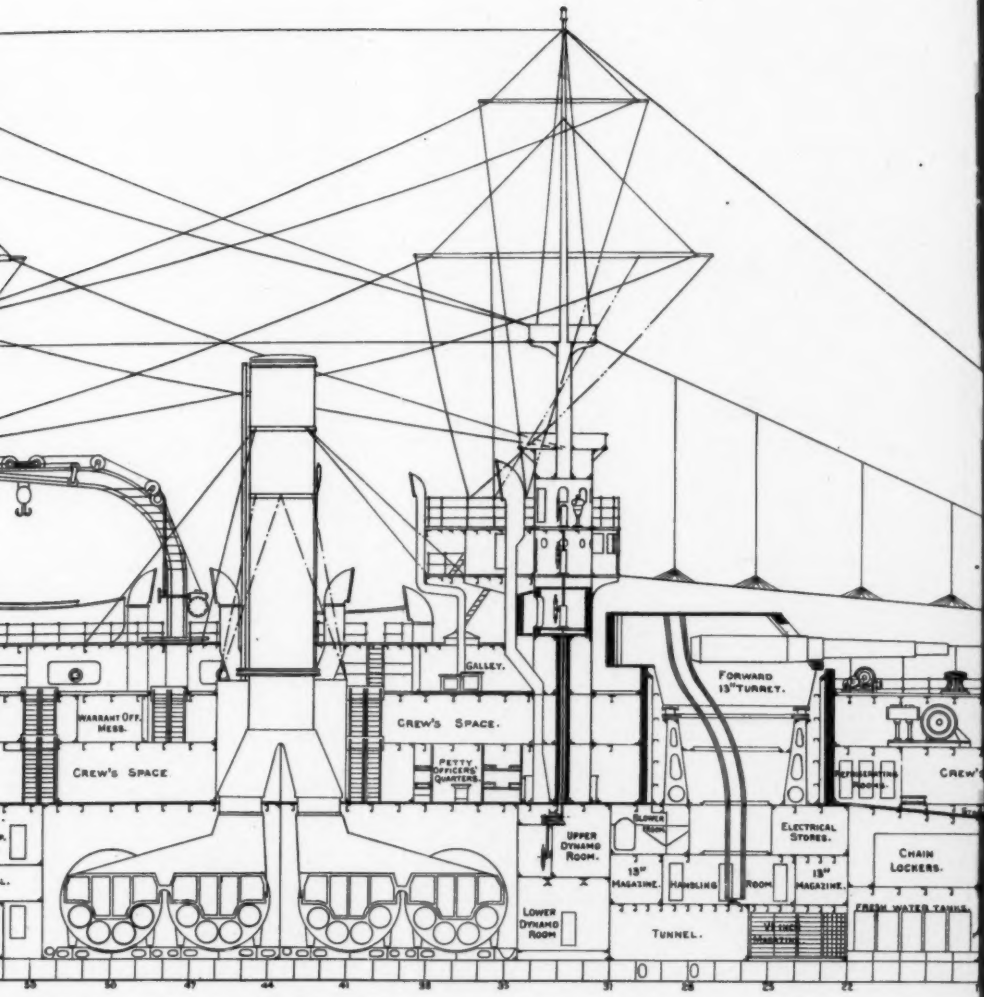
One tank, gallons	450.24
Twenty tanks, gallons	9,004.80

FIRE HYDRANTS.

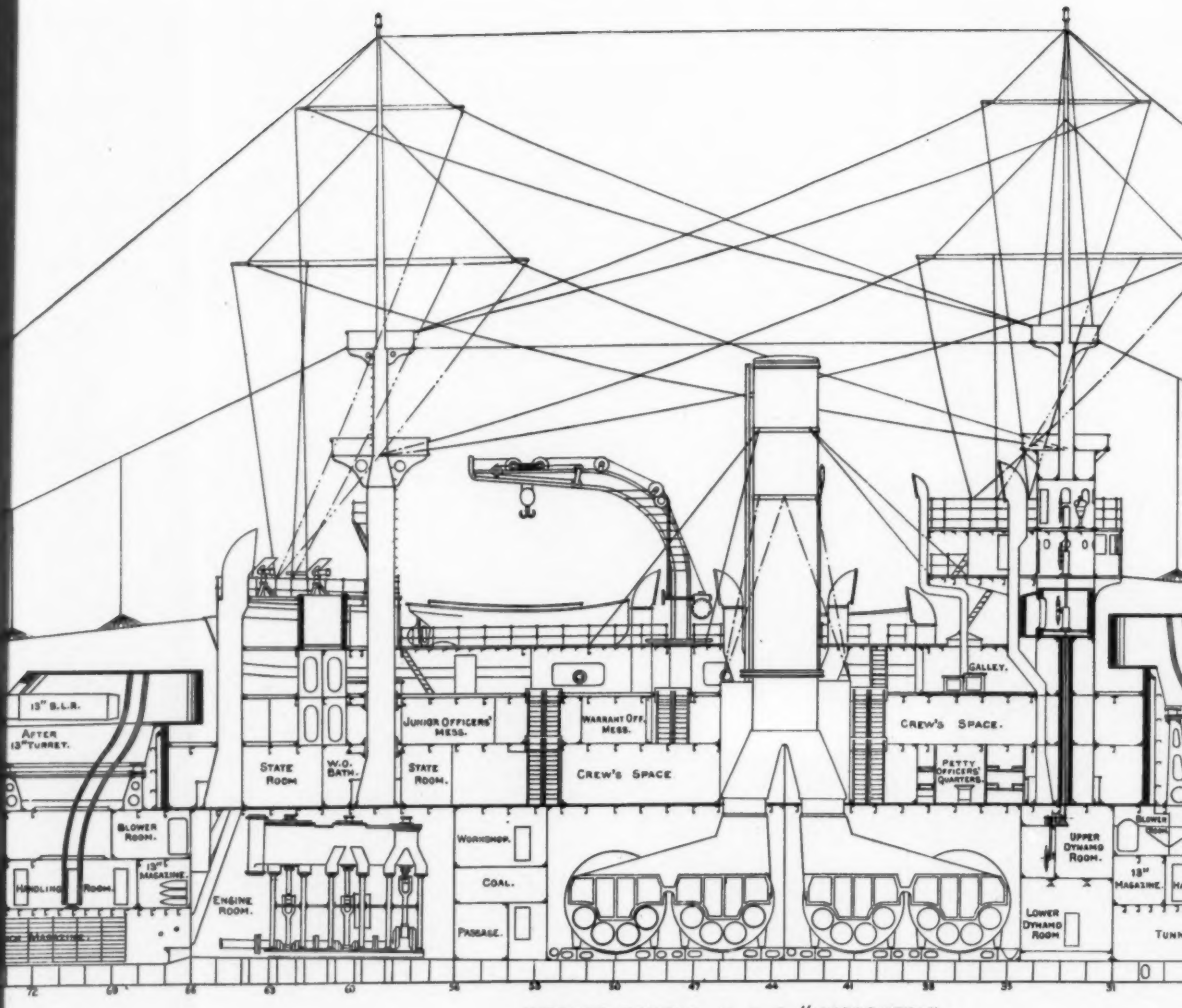
Upper deck	7
Main deck	8
Berth deck	14
Splinter deck	5
Platform deck	4
Engine and boiler rooms	10


13" B.L.R.
AFTER 13" TURRET.
6 PDR
AFTER CABIN
PASSAGE.
ADMIRAL'S CABIN.
WARD ROOM.
ADMIRAL'S STORES.
ADMIRAL'S
6 PDR
STEERING GEAR.
FRESH WATER TANK.
TRIMMING TANK.
BREAD AND DRY PROVISIONS.
HANDLING
ROOM.
13" MAGAZINE.
BLOWER ROOM.
WINCH MACHINE.
ENGINE ROOM.
STATE ROOM
W.C. BATH.
STATE ROOM.
JUNIOR OFFICERS' MESS.
WORKSHOP.
COAL.
PASSAGE.
85
80
75
72
69
66
63
60
54
39

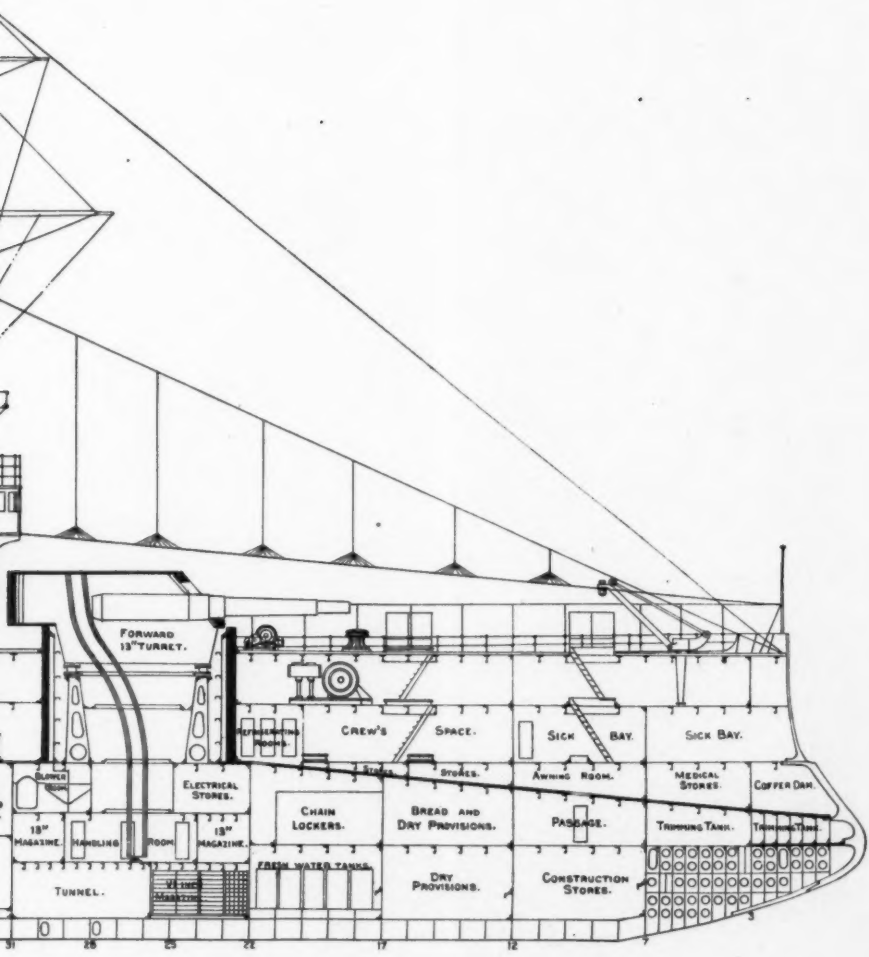
INBOA



INBOARD PROFILE, U. S. S. "WISCONSIN."



INBOARD PROFILE, U. S. S. "WISCONSIN."


FORWARD 13" TURRET.
CREW'S SPACE.
SICK BAY.
SICK BAY.
STORES.
STORES.
AWNING ROOM.
MEDICAL STORES.
COFFER DAM.
BLOWER ROOM.
ELECTRICAL STORES.
CHAIN LOCKERS.
BREAD AND DRY PROVISIONS.
PASSAGE.
TRIMMING TANK.
TRIMMING TANK.
13" MAGAZINE.
HANDLING ROOM.
13" MAGAZINE.
FRESH WATER TANKS.
TUNNEL.
DRY PROVISIONS.
CONSTRUCTION STORES.
31
26
23
22
17
12
7
3

SMALL BOATS.

No.	Kind.	Men.	Water and provisions.
			Pounds.
1	36-foot steam cutter	50	500
1	33-foot steam cutter	42	420
1	33-foot sailing launch	60	600
6	30-foot cutters, each	40	400
1	30-foot barge	40	400
2	30-foot whale boats, each	28	280
1	30-foot gig whale boat	18	180
2	20-foot dinghies, each	12	120
2	18-foot balsas	...	.....
2	12-foot punts	...	.....

COMPLEMENT.

Admiral,	1
Captain,	1
Ward-room officers (state rooms),	15
Junior officers (berths),	10
Warrant officers (berths),	7
Seamen's branch,	283
Artificers' branch,	19
Engine-room force,	124
Special branch,	12
Messmen,	23
Additional for flagship,	35
Marines,	60
	590

Recapitulation.

Officers,	34
Crew,	496
Marines,	60
Total,	590

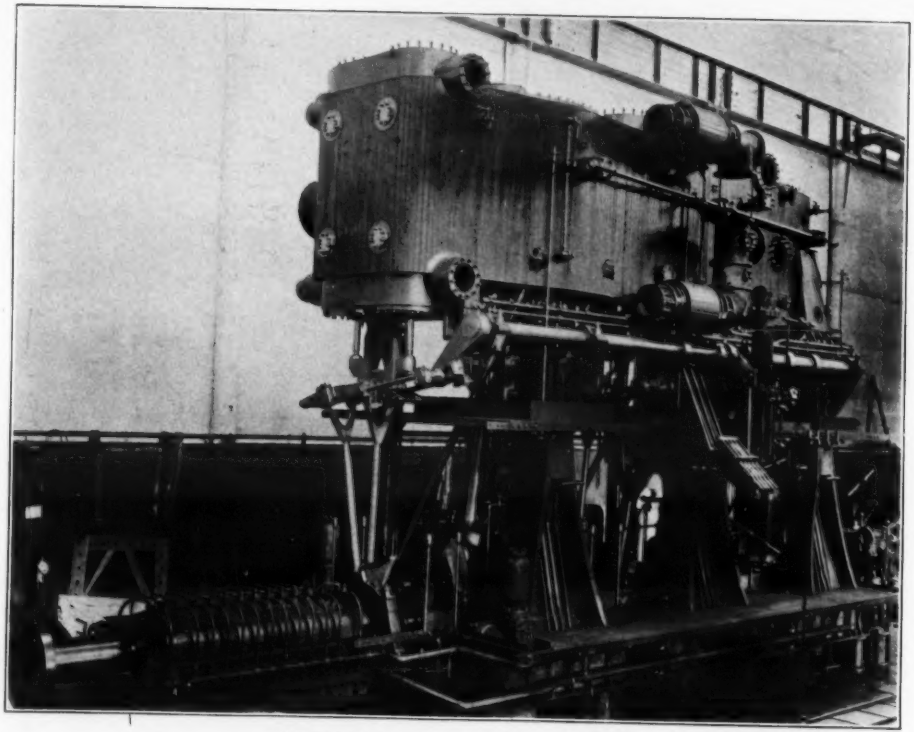
BOILER DATA.

Number	8
Type	Single-ended Scotch.
Length, feet and inches	9–11¼
Diameter, outside, feet and inches	15– 8

Thickness of shell, inches	$1\frac{7}{16}$
heads, top, inches	$1\frac{7}{16}$
bottom, inch	$\frac{3}{4}$
tube sheets, inch	front, $\frac{7}{8}$, back $\frac{3}{4}$
furnaces, inch	$\frac{9}{16}$
Combustion chambers, number in each boiler	2
thickness of plates, inch	$\frac{9}{16}$
depth at top, inches	$27\frac{1}{2}$
Furnaces, greatest internal diameter, feet and inches	3– $5\frac{7}{8}$
least internal diameter, feet and inches	3– 3
length of grate, feet and inches	6– $1\frac{7}{8}$
number in each boiler (corrugated)	4
Tubes, outside diameter, inches	$2\frac{1}{2}$
length between tube sheets, feet and inches	6– $6\frac{1}{16}$
number of ordinary, each boiler	388
number of stay, each boiler	154
spaced vertically, inches	$3\frac{1}{2}$
horizontally, inches	$3\frac{3}{4}$
thickness of ordinary, B.W.G., No. 9, inch	.148
stay, B.W.G., No. 6, inch	.203
Diameter of rivets in shell sheets, inches	$1\frac{5}{16}$
screw stays, inches	$1\frac{1}{2}$
Number and diameter of through upper braces, inches	10 of $2\frac{3}{4}$
lower braces, inches	2 of $1\frac{3}{4}$
braces from head to back tube sheets, inches	15 of $1\frac{3}{4}$
Heating surface, tube, square feet, each boiler	2,307.65
plate, square feet, each boiler	343
total, square feet, each boiler	2,650.65
Grate surface, square feet, each boiler	80
Area through tubes, square feet	12.813
Funnel, 9 feet 6 inches by 6 feet 6 inches, ellipse area, square feet, each,	48.5
Height of funnel above lower grates, feet	86.5

ALL BOILERS.

Heating surface, tube, square feet	18,461.2
plate, square feet	2,744
total, square feet	21,205.2
Grate surface, square feet	640
Area through tubes, square feet	102.504
Ratios:	
Tube H.S. to G.S.	28.84
Plate H.S. to G.S.	4.29
Total H.S. to G.S.	33.13
Area through tubes to G.S.	.16



PORT ENGINE, U. S. S. "WISCONSIN."

MAIN ENGINE DATA.

Number of engines	2
Cylinders, number for each engine	3
diameter of H.P., inches	33½
I.P., inches	51
L.P., inches	78
Stroke of pistons, inches	48
Valves, diameter of H.P. (one for each cylinder), inches....top,	16½
I.P. (two for each cylinder), inches....top,	21¾
L.P. (four for each cylinder), inches....top,	18
H.P., inches....bottom,	15
I.P., inches....bottom,	20½
L.P., inches....bottom,	20½
Valve stems, H.P. (1), diameter, inches	2¾
through valve, inches	2
I.P. (2), diameter, inches	2¾
through valve, inches	2
L.P. (4), diameter, inches	2¾
through valve, inches	2
Main steam pipe, diameter, inches	12
Exhaust pipe to I.P. cylinder, (1) end, diameter, inches	15½
L.P. cylinder, 15 inches I.P. end, (8) 11 inches	——
end, inches	——
condenser (2), diameter, inches	20½
Volume swept by H.P. piston (mean), per stroke, cubic feet	23.84
I.P. piston (mean), per stroke, cubic feet	56.09
L.P. piston (mean), per stroke, cubic feet	132.04
Net area of I.P. to H.P. piston	2.34
L.P. to I.P. piston	2.36
L.P. to H.P. piston	5.53
Clearance of H.P. cylinders, per cent....top, 18.9; bottom,	22.8
I.P. cylinders, per cent....top, 18.9; bottom,	24.7
L.P. cylinders, per cent....top, 17.5; bottom,	22.9
Piston rods, diameter, inches....outside, 7⅛; inside,	1½
Connecting rods, length from center to center, feet	8
diameter of upper end, inches	7½
lower end, inches	8½
inside, inches	4½
crosshead bolts (2), diameter, inches	4¼
crank-pin bolts (2), diameter, inches	4¼
Crossheads, surface (ahead), square inches	459
(backing), square inches	357
pins, diameter, inches	9½
length, inches	9
Reversing gear, steam cylinder, diameter, inches	14
controlling cylinder, diameter, inches	7½

Crank shafts, diameter, inches	14¾
axial hole, inches	7½
coupling discs, diameter, inches	29
thickness, inches	3½
bolts in one flange (6), diameter, inches	3½
journals, diameter, inches	14¾
length, inches	16½
length of each section, feet and inches	8–10
Crank pins, diameter, inches	14¾
length, inches	17
axial hole, inches	7½
Crank webs, width, inches	16½
thickness, inches	10
Thrust shafts, diameter, inches	14
of coupling disc, inches	29
axial hole, inches	9
collars, number, each shaft	11
diameter, inches	21½
thickness, inches	2
distance between, inches	4
surface, total for both engines, square inches	4,600.64
Line shafts, diameter, inches	14
axial hole, inches	9
diameter of coupling discs, inches	29
length, feet and inches	36– 3
Propeller shafts, diameter, inches	14¾
axial hole, inches	9¾
length, feet and inches	53–10½

PROPELLER DATA.

Type: Three-bladed, expanding pitch, adjustable from 16 feet 6 inches to 18 feet 6 inches. Top of blade thrown back 6 inches.

Material: Manganese-bronze.

Number of blades	3
Diameter, feet and inches	15– 6
of hub, feet and inches	4– 2
Length of hub, feet and inches	3– 4
Pitch as set, entering edge, feet and inches	16– 6
leaving edge, feet and inches	18– 6
mean, feet and inches	17– 6
Greatest width of blade, feet and inches	4– 9
Helicoidal area of each screw, square feet	68
Disc area, square feet	188.7
Pitch ÷ diameter	113
Center of hub above keel, feet and inches	8–10⅝
Top of propeller, immersed, feet and inches	6–11

Shafts incline up at angle	1° 15′ 36.69″
diverge at angle	1° 27′ 3.37″
Center of hub forward of aft perpendicular, feet and inches	15– 1
from middle line, feet and inches	13– 1¾
Immersed midship section at mean draught of 23 feet 6 inches, square feet	1,625
Ratio developed blade area to immersed midship section	1 to 11.21

While the machinery of the *Wisconsin* is identical, as already stated, with that of other vessels of the same class, there are some peculiarities in design that are noteworthy:

(1) The H.P. valves of the main engines are double ported on the steam side, to reduce wire drawing.

(2) The main-engine bed plates are of cast bronze.

(3) The back frames of the main engines are built up Y frames of forged steel, the crosshead guide of cast iron being secured between two steel forgings by bolts. The use of this rigid framing involves a very slight increase in weight (stated to be but 500 pounds per cylinder).

(4) All pipe flanges are faced in a special flange facing machine *after* they are attached to the pipe. The joints in all steam piping are metallic.

(5) The valves of the main engines are balanced by varying the diameters of the upper and lower ends of the valves.

THE TRIAL.

The official full-speed trial of the *Wisconsin* took place on Thursday, October 11, 1900, over the regular Santa Barbara Channel course, a fraction over 32 knots in length. The trial consisted of two runs in opposite directions over this course. On the first run, with a head swell and with a moderate breeze from within one point of astern, the speed was 17.42 knots. On the return run, with a head wind and with the swell from astern, the speed fell to 16.928 knots, the revolutions of the engines being almost the same for the two runs. During the time between the two runs there was no change in the relative speed of the two main engines, but both were slowed down about four turns per minute without change in adjustment of throttle or valve gear.

AUXILIARY STEAM MACHINERY.

No.	Class.	Type.	Location.	Size.	Remarks.
2	Main air pumps	Dow, vertical, beam, single-acting	1 in each engine room	2, 12″ steam, 23″ water, 20″ str.	Pump valves, disc, bronze.
2	Main circulating pumps	Centrifugal double inlet	do	Engine, 8½″×13½″×8″	Runner, 36″, suct., 17″, discharge, 17″.
2	Aux. air and cir. pumps	Dow, vertical, beam	do	8″ steam, 8″ water, 12″ air; 10″ stroke.	1 steam cylinder above circ. pump.
2	Main feed pumps	Dow, vertical, duplex	do	12″ steam, 7¼″ water, 12″ stroke	Metallic pump valves.
4	Auxiliary feed pumps	do	1 in each fire room	do	Do.
2	Water-service pumps	do	1 in each engine room	do	Rubber pump valves.
2	Fire and bilge pumps	do	do	do	Do.
8	Forced-draft blowers	Vertical, inverted, compound, direct-acting. Cranks at 180°.	2 in each fire room	5″×8″×6″ stroke	Fans, 60″ diam. Area, induction, 1,018 sq. ins.; discharge, 2,000 sq. ins.
6	Hull ventilating blowers	do	2 splinter deck forward, 2 splinter deck aft, 2 machine shop.	4½″×8″×4″ stroke	Fans, 57″ diam. Area, induction, 707 sq. ins.; discharge, 798 sq. ins.
2	Engine room do	do	1 for each engine room	do	Do.
8	Dynamo engines	4 Gen. Electric design for turrets. 4 Union Iron Works design.	Dynamo room	8½″×13½″×8″ stroke	Capacity, 400 am. at 82 volts, 400 revs.
2	Hyd.-pump steering gear	Dow, vertical, duplex, differential	1 in each engine room	Steam 2, 10″, water, 3″ and 4¼″; stroke, 12″.	Metallic valves.
1	Distiller circulating pump	Dow, vertical, double-acting	In No. 1 fire room	6½″ steam, 6″ water, 10″ stroke	Rubber valves.
1	Fresh-water pump	do	On berth deck	5½″ steam, 5½″ water, 8″ stroke	Do.
1	Drain-tank pump	Dow, horizontal, double-acting	In fire room No. 2	do	Do.
2	Brine and fresh-water evaporator pumps.	do	1 in each engine room	4½″ steam, 2¾″ brine or water, 6″ stroke.	Do.
1	Feed and fresh-water evaporator pump.	do	In drum room	do	Do.
2	Hot-water drainage evaporator pumps.	do	1 in each lower dynamo room	2″ steam, 1¼″ water, 2¾″ stroke.	Metallic valves.
2	Ice machines	Allen, dense-air, 1-ton	Refrigerating room, berth deck.	9″ steam, 5¾″ compr., 4¾″ exp., 1¾″ circ., 1¾″ primer, 10″ str.	
4	Ash hoists	Horizontal, simple	1 for each fire room	4″×6″ stroke	31-foot hoist of bucket.
1	Anchor engine	Hyde, vertical, 2-cylinder, simple	Main deck, forward	2, 15″×14″	
5	Winches	Hyde, horizontal, 2-cyl., geared	Upper deck	2, 8″×8″	
2	Air compressors	Whitworth, vertical	1 in each compressor room	8¾″ steam, 4″ stroke	Triple, compound, compressor cylinder, 3 sets on each engine.
1	Workshop engine	Vertical, inverted, simple	In workshop	6″×5″	
2	Boat-crane engines	Double, vertical, geared	On crane platforms	2, 8″×8″ stroke	10,000-pound hoist, 28-foot reach, 40-foot lift per minute.
2	Turning engines	Vertical, simple, geared	1 attached to each main engine	6″×6″	
2	Reversing engines	Steam and oil, direct-acting	do	14″ steam, 7¾″ oil, 20¼″ stroke	
2	Bilge and sanitary pumps	Eccentric-driven, horizontal, single-acting.	do	Each 7¼″×6″ stroke	Metallic valves.

No live steam was admitted to the receivers during the trial. During the first hour of the trial, the throttle valve was opened from .7 to .8. No other change was made in throttle or valve gear during the trial.

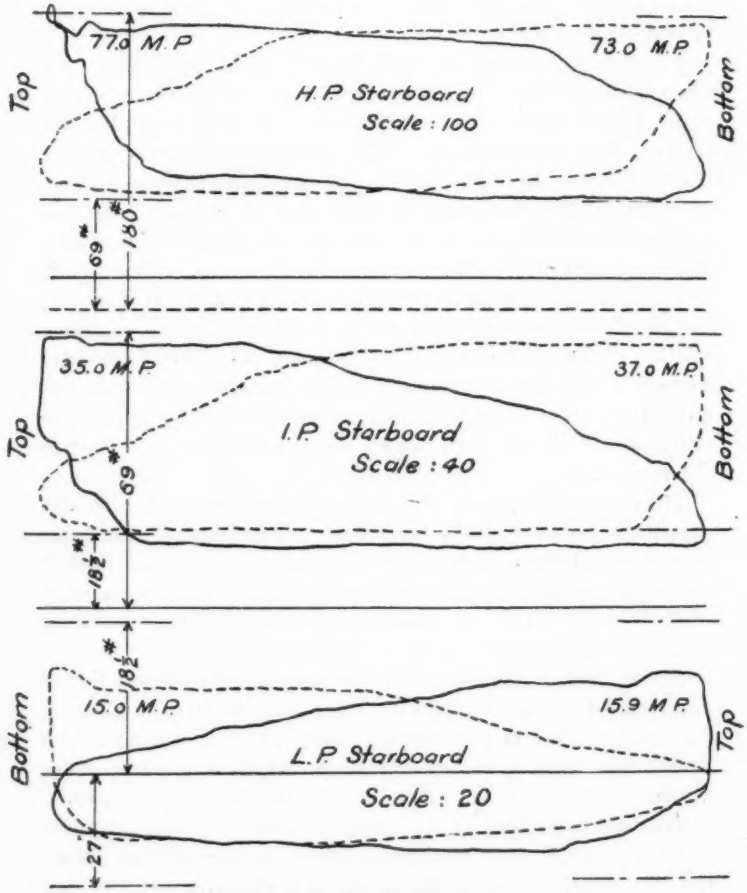
The forced-draft system worked with great regularity. An effort was made to keep the air pressure in the fire rooms at about $\frac{7}{8}$ inch of water. Whenever this figure was exceeded the blowers were slowed down. The average air pressure during the trial was 0.924 inch, less than the one inch pressure allowed. In spite of this fact and in spite also of the fact that one evaporator was run during the trial and furnished all necessary make-up feed water, the safety valves of the boilers lifted frequently. This desirable result was achieved by: (1) Carrying the fires comparatively thin on the grates; (2) The excellent quality of fuel used, and (3) The efficient use of the feed-water heaters in which was condensed all the exhaust steam from the auxiliaries other than the main air pumps and by which the temperature of the feed water was raised over 60 degrees Fahrenheit.

Throughout the trial all the machinery worked with the greatest regularity and smoothness. There was no excessive vibration and there were no hot bearings. As a precautionary measure, a spray of water was used from time to time on some of the journals. The usual careful inspection of all the machinery after the trial showed it to be in excellent condition.

Before the trial the *Wisconsin* had made a trip to Puget Sound and return from San Francisco. No defects developed. During the trial no defects developed. After the trial, on the return from Santa Barbara to San Francisco, the average speed under *natural draft* was 15.292 knots. Again no defects developed.

That the *Wisconsin* is an "easy steamer" is shown by this natural-draft run, and this is emphasized by the fact that during the trial, with the safety valves lifting frequently and the machinery working at 26 per cent. above its designed capacity, no firing tools were used except shovels.

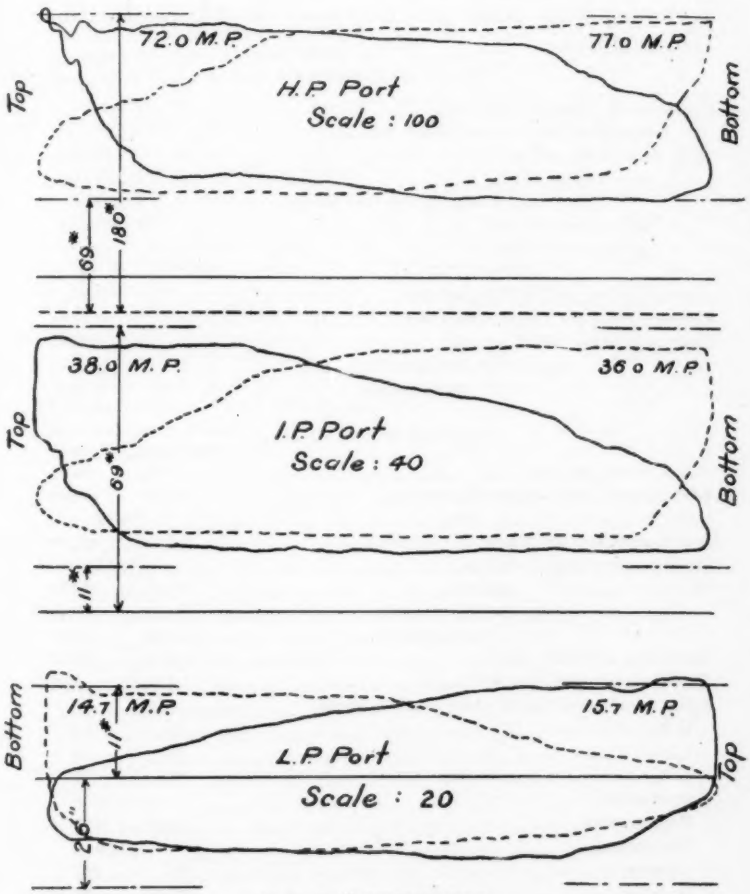
The coal was carefully measured and the coal account is believed to be accurate. For conditions such as obtain on a



U. S. S. "WISCONSIN."

AVERAGE CARDS—STARBOARD ENGINE. FULL SPEED TRIAL OCTOBER 11, 1900.

Data page 996.



U. S. S. "WISCONSIN."

AVERAGE CARDS—PORT ENGINE. FULL SPEED TRIAL OCTOBER 11, 1900.

Data page 996.

forced-draft trial the result of 1.8882 pounds of coal per I.H.P. per hour is gratifying to designers and builders.

The following tables give the results of this exceptionally successful trial.

CONTRACT-TRIAL DATA.

Displacement (mean on trial), tons	11,500
Area immersed midship section, square feet	1,619.6
Mean speed, knots per hour	17.174
I.H.P., main engines only	12,321.695
air, circulating and feed pumps	12,452.36
all machinery in use	12,608.992
per square foot G.S.	19.7016
H.S	0.59462
Heating surface per I.H.P., square feet	1.68175
Cooling surface per I.H.P., square feet	1.2374
Coal per hour per I.H.P., pounds	1.8882
square foot, G.S., pounds	37.2
H.S., pounds	1.123

SYNOPSIS OF STEAM LOG.

	Starboard.	*Port.*
Slip of screws, per cent	16.63	16.73
Revolutions of main engines per minute	119.15	119.28
mean	119.22	
Piston speed, feet per minute	953.2	954.24
Steam pressure, gage, at boilers	187.15	
engines	183.6	182.83
absolute, in first receiver	84.67	80.44
second receiver	32.57	26,53
Vacuum in condenser, inches	26.73	26.08
Throttle-valve opening in tenths	7.8	7.94
Steam cut-off in fraction of stroke from beginning:		
H.P.	71	71
I.P	70	71
L.P.	71	70
Temperatures:		
Engine rooms	94.87	101.88
Fire rooms	94.79	
Uptakes	667	797.5
Injection	58.88	64.44
Discharge	112.06	122
Air-pump discharge	100.13	100
Feed water	166.33	159.66
Air pressure in fire rooms, inches of water	0.924	

Mean effective pressure:		
H.P.	74.062	77.8
I.P.	37.33	36.555
L.P.	15.31	15.53
Aggregate equivalent pressure referred to L.P.	44.495	45.11
Revolutions or double strokes per minute:		
Main air pump	37.98	20.29
Main circulating pump	175.14	163.59
Main feed pump	49.07	40.1
Auxiliary air and circulating pump	55	48
Bilge pumps	49	49
W. S. pump	43	
Sanitary pumps	40	
Hydraulic pump	15	
Ice machine	112.5	
Evaporator feed pump	15	
Average 8 F. D. blowers	450	
Average 2 E. R. blowers	450	
Average 6 B. D. blowers	316.5	
Dynamos, mean total load, 80 volts, ampères	356.8	
Indicated Horse Power	*Starboard.*	*Port.*
Main engines, H.P.	1,837.745	1,931.65
I.P.	2,177.5	2,135.6
L.P.	2,103.3	2,135.9
Totals	6,118.545	6,203.15
Air pumps	9.277	5.256
Circulating pumps	13.753	12.169
Feed pumps	47.34	42.87
Bilge pumps	6.094	
Evaporator, estimated	10	
Evaporator feed pump, estimated	0.4	
Hydraulic pump, estimated	6	
Ice machine, estimated	10	
Water-service pump	7.95	
Sanitary pump	7.95	
B. D. blowers	12.42	
E. R. blowers	6.50	
F. R. blowers	42.848	
Dynamos	42.116	
All machinery in operation during trial	12,608.992	
Coal:		
Hand picked	Harris Cardiff.	
Pounds per hour	23,808	
Pounds per hour per I.H.P.:		
All machinery in operation during trial	1.8882	
Main engines, air, circulating and feed pumps	1.912	

GLASS REVEALER FOR STUDYING CONDENSATION IN STEAM-ENGINE CYLINDERS AND RENDERING THE EFFECTS VISIBLE.*

Some years ago the writer introduced a small glass instrument, named by Hirn a "revealer," for studying the condensation of steam in an engine cylinder. Since then some improvements have been made and the following results obtained: A revealer was placed on the top of both the high and low-pressure cylinders of a compound steam-jacketed, vertical, condensing beam engine, working in a factory at a speed of about 35 revolutions per minute. The diameter of the low-pressure cylinder is 26 inches, with 3-foot stroke, and that of the high-pressure cylinder, $7\frac{5}{16}$ inches, with 14-inch stroke.

GENERAL DESCRIPTION.

Each revealer has two cylindrical glasses—Fig. 2—fixed concentrically one inside the other, thus forming a hot-air jacket for the inner glass, with an annular space of 10 mm. (0.40 inch) between them. The inner glass cylinder is smaller in diameter than formerly, the connecting pipe to the motor cylinder is straight and the cock and pipe much larger, so that the connection is more direct. Inside the inner glass is a small cast-iron cylinder. All the external and glass surfaces are well covered with removable, thick, non-conducting material to diminish radiation. The object of the instrument is to reproduce as nearly as possible, and render visible, what takes place inside the steam cylinder, and is necessarily concealed. No vessel, such as a revealer, exterior to the cylinder, can be under exactly the same conditions as the internal parts, because the cylindrical portion of the latter is cleaned by the piston at every half revolution,

* Abstract of paper by Bryan Donkin before the Institution of Mechanical Engineers, London. From "The Iron Age."

IMPROVED GLASS REVEALER

Fixed on Steam-Engine Cylinder for showing Effects of Condensation, Re-evaporation, etc.

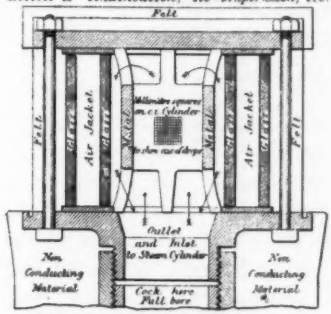
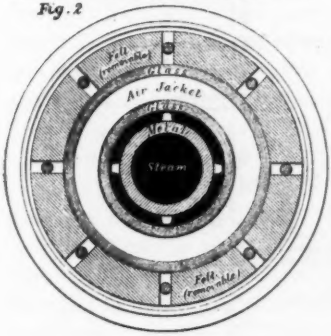


Fig. 2



IMPROVED GLASS REVEALER.

With two Cylindrical Glass Cylinders and Internal Steam or Water Jacketed Cylinder

Fig 3

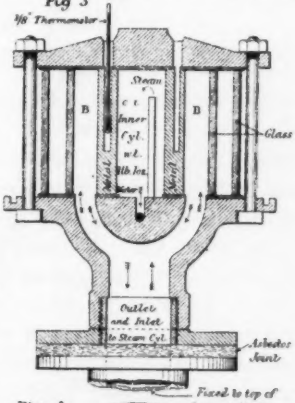


Fig. 4

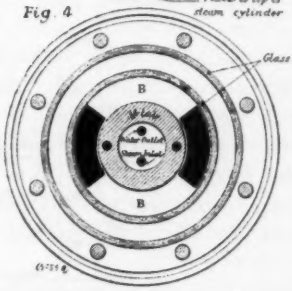
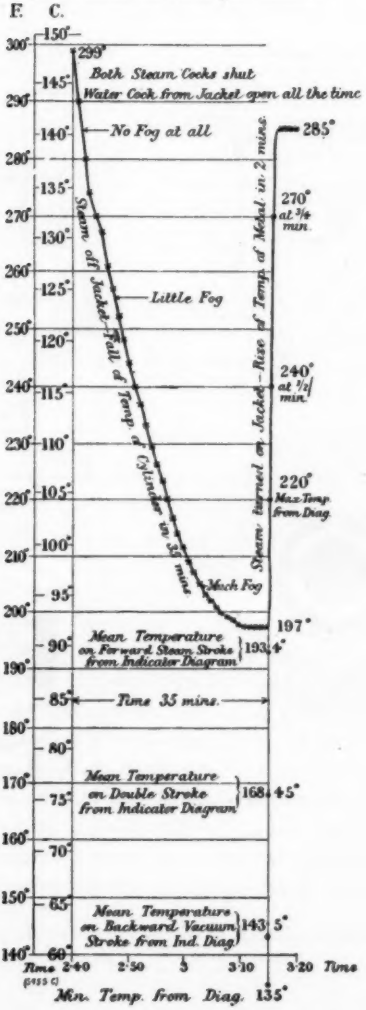


Fig. 5. Curve showing Fall of Temperature of Cyl. shown in Fig 3 & 4. exposed to Exh. Stroke &c. and so made as to be jacketed or not at pleasure, fixed on Low-Press. Cyl. of Factory Engine. Speed about 35 revs. per min.



and the clearance surfaces are greasy. Nevertheless the effects of condensation, as shown by the revealer, may be within 85 to 95 per cent. of the truth, and all results can be accurately seen, magnified and studied. In all steam cylinders condensation takes place in the internal walls when working with saturated steam. In this revealer, under the above conditions, very fine globules of water, the results of condensation, will be seen close together, the appearance of which the author will first endeavor to describe for the low-pressure cylinder.

REVEALER ON THE LOW-PRESSURE CYLINDER.

The modifications of the improved revealer have caused a great reduction in the size of the minute globules on the glass walls as compared with those shown in former experiments. After the revealer has become comparatively warm, and when the steam is first admitted into it from the low-pressure cylinder, the globules, from 1 to 2 mm. (0.04 to 0.08 inch) in size, are much larger than those seen after working about 10 to 15 minutes, when all the parts have had time to acquire a normal temperature. Commencing with the steam stroke, the first result observed when the steam is admitted to the revealer is a light and not very dense cloud or white fog coming up from the engine cylinder. This is precipitated upon the hot internal glass and cast-iron walls, and is seen as very fine globules or particles of moisture or condensation. This cloud is sufficient to obscure a light placed on the opposite side of the revealer. Re-evaporation then rapidly follows before the end of the steam stroke. On the opening of the exhaust valve the cloud passes quickly off into the cylinder and the glasses become quite transparent. A good illumination renders the incoming and outgoing clouds of mist quite visible.

Each time the steam enters, a large number of very small globules, at about equal distances from each other, are immediately seen upon the glass walls. These are from $\frac{1}{20}$ to $\frac{1}{25}$ mm. (0.002 to 0.0016 inch) in diameter, or less, and are distributed in a fairly uniform manner over the surfaces. The time occupied by the deposition of the globules and their re-evaporation is, perhaps,

one-tenth second. Before the end of the steam stroke, the duration of which is about one-half second, and before the exhaust valve opens, the smaller globules, $\frac{1}{25}$ mm. in diameter, are all evaporated. During the exhaust stroke the larger globules are also re-evaporated, but not so quickly. The internal surfaces are sometimes quite dry and at other times a few drops of moisture remain on them.

To facilitate the study of these minute condensed globules or wet particles on the inside of the inner glass a microscope is used, and the glass and iron walls well illuminated. It is interesting to watch not only the formation of the globules, but also their rapid disappearance by evaporation. Any change in the working conditions of the engines, by altering the expansion or speed, is at once detected. The globules change in diameter, becoming larger or smaller than before, probably as the difference of temperature between the walls and the steam is increased or diminished. Very soon, however, the glass walls take up a certain temperature and constant conditions again prevail, differing only in size of the globules from those previously seen. By artificially cooling the outside of the short iron connecting pipe between the glass revealer and the cylinder with cold water and a sponge the fog or cloud on admission of the steam stroke is much increased, as is also the diameter of the globules on the internal glass walls. The steam seems to be chilled.

Two other revealers, with single glasses and longer and uncovered connecting pipes, fixed on the same end of the same low-pressure cylinder, gave during the same stroke greater condensation and larger globules, with thicker fog inside. This was the result which might be expected.

CONDENSATION ON THE CAST IRON FROM THE LOW-PRESSURE CYLINDER.

Inside the inner glass cylinder or revealer a small cast-iron cylinder was placed, to observe the condensation produced on this metal and to measure the size of the globules. Vertical and horizontal cross lines 1 mm. apart were marked on this cylinder, forming small areas of 1 sq. mm., Fig. 2. These squares, being visible under the microscope, afford a useful

means of estimating the size of the minute globules. The condensation observed was practically the same on the cast-iron as on the inner glass cylinder, both being probably at the same temperature and touched by the same steam.

The various phenomena may be summarized as follows, after the normal conditions are established, for the low pressure cylinder:

ONE REVOLUTION.

1 Fog rising from cylinder into revealer.....	Steam stroke, duration about ½ second.
2. Deposition of moisture and water on glass and metal surfaces..	
3. Rapid re-evaporation during expansion.....	
4. Exhaust opens. Fog sinking back to cylinder.....	Exhaust stroke. duration ½ second.
5. Glasses quite clear. Surface nearly dry.....	

Internal diameter of glasses, 67 and 44 mm. (2.64 and 1.73 inches). External diameter of cast-iron cylinder, 35 mm. (1.38 inches). Height of glasses and cylinder, 57 mm. (2.24 inches).

The pressure in the low-pressure cylinder varies a little at different times of the day; the maximum initial is generally about at atmospheric pressure. At times in the revealer no condensation is visible and no re-evaporation, only very slight fog in and out.

REVEALER ON THE HIGH-PRESSURE CYLINDER.

A second revealer, with double glasses and a small cast-iron cylinder in the center, was placed on the top of the high-pressure cylinder and well protected from heat losses. All the general series of effects were the same as those on the low-pressure cylinder during one revolution. The only marked difference between the two cylinders was in the size of the condensed globules on the glass and cast-iron surfaces. These were about $\frac{1}{4}$ to $\frac{1}{5}$ mm. (0.01 to 0.008 inch), some few being $\frac{1}{2}$ mm. (0.02 inch) in diameter. This means more condensation in this cylinder than in the low-pressure, which agrees with calculation and experiment. This cylinder is jacketed with live steam at a pressure of 50 pounds, or about four atmospheres. With the low-pressure cylinder the pressure is the same in the jackets—namely, four atmospheres—and as the mean working pressure is much lower

than in the high-pressure cylinder, there is a much greater difference of pressure and temperature between cylinder and jacket to keep the low-pressure cylinder walls hot. The jacket is more efficient on the low-pressure cylinder and less condensation is to be expected. This agrees with experimental results. In this plan the two revealers can be seen and compared, both working at the same time on the top of each cylinder.

To make sure that the varying pressures of steam in the revealers were really the same as in the engine cylinder two indicators were fixed, one on the top of the revealer and the other on the top of the cylinder, and diagrams taken during the same stroke. The diagrams were exactly the same from the revealer as from the cylinder.

The range of temperature in the high-pressure cylinder is generally about 101 degrees Fahrenheit and that in the low-pressure cylinder some 85 degrees Fahrenheit, but the pressures and temperatures vary somewhat with the work. This double-glass revealer forms a kind of transparent prolongation of the steam-engine cylinder.

One result of these observations is to prove that there cannot be any thin film of condensation water on the internal surface of steam-engine cylinders, whether the cylinder is relatively hot or cold, as there has so often been supposed to be. The difference of temperature between the walls and the working steam seems to affect only the size of the globules of water.

It is very difficult for the eye to follow the rapid and violent movements of fog and small globules of water in all directions on the internal surfaces, and also their very quick and violent disappearance by re-evaporation at exhaust.

It is now generally admitted that condensation of steam in engine cylinders is caused chiefly by the difference of temperature between the internal surfaces and the steam in contact with them, which it should be the object of engineers to minimize. Condensation is reduced in quick-revolution engines and where dry or superheated steam is used. It assumes various forms, water particles, fog or globules of different diameters. These seem always to be in rapid motion and not stagnant. On admission the

internal metal walls are heated by the steam. Other heat changes occur in each revolution, tending to equalize the temperature between the solid iron, the liquid water and the gaseous steam.

STEAM REVEALER WITH TWO CYLINDRICAL GLASSES, AND INTERNAL STEAM OR WATER-JACKETED CAST-IRON CYLINDER IN CENTER.

The following further experiments have been made: A small cast-iron cylinder, Figs. 3 and 4, is fixed in the center of the revealer, and so arranged that boiler steam can be introduced inside it to raise its temperature, provision being made to drain the water away. Cold water can also be passed through it continuously to reduce its temperature. The mean temperature of this inner cylinder is taken by means of a mercurial thermometer about 2½ mm. (0.1 inch) in diameter, placed in a hole in the center of the wall. The steam from the engine cylinder passes at each stroke into and out of the annular space *B B* between the cylinder and the inner glass. The fog or condensation on the walls can be observed through the glass cylinder by the eye or microscope. In this way the cylinder can be either heated by steam or cooled by water, and the fog effects noted for different temperatures of the small cylinder inside the revealer. To get the mean temperature of steam in the steam cylinder indicator diagrams are taken and the mean temperature calculated from the pressures. Thus, when all the conditions are constant, the temperatures attained by the inner cylinder can at any time be compared with the mean temperature of steam touching its walls.

EXPERIMENT.

The revealer shown in Fig. 3 was fixed on the top of the low-pressure jacketed cylinder of the engine and placed in communication with it. The speed of the engine was about 35 revolutions per minute. Boiler steam was admitted into the inside of the small central cylinder, and the revealer was kept going with the power on the engine constant for a given time until all the working conditions were normal. At a given moment, with the wall at 299 degrees Fahrenheit, the steam was shut off from the jacket and notes were made of the gradual decrease of the

wall temperature, arising, doubtless, from the exposure of the hotter walls to the cooler steam stroke, and to the still cooler exhaust stroke to condenser. This had already been noticed in a preliminary experiment. The wall temperature was read every minute until a minimum of 197 degrees Fahrenheit, was reached and maintained. The steam was then turned again into the jacket and the rapid rise of the wall temperature noted up to 285 degrees Fahrenheit, in about one minute. In 35 minutes the temperature of the cylinder decreased from 299 to 197 degrees, equal to 102 degrees Fahrenheit, about 2.9 degrees Fahrenheit per minute. The mean temperature of the steam on double stroke was 168.45 degrees Fahrenheit, or 28.5 degrees Fahrenheit cooler than the mean temperature of the small cylinder walls, 197 degrees Fahrenheit. From the weight of the small cylinder, 1.06 pounds, its loss in thermal units per minute can be calculated. This amounts to 0.384 thermal unit per minute for 0.095 square foot, and is equal to four thermal units per square foot per minute lost by the walls to the condenser. In the above experiment there was very little radiation from the revealer, as it was just above a steam jacketed and hot cylinder; and also very little conduction of heat along the connecting pipe, which is well covered, as is also the revealer.

CONDENSATION OF FOG IN THE REVEALER.

With steam in the jacket no fog is visible, nor can any condensation on the walls be detected by the eye. They are much hotter than the incoming steam. About one-quarter of an hour after shutting off the steam in the jacket a little fog is visible, and at the end of thirty-five minutes much fog is seen and slight condensation on the walls; in other words, as the temperature of the walls is gradually reduced, the amount of fog and condensation increased.

LATE EXPERIMENTS.

In late experiments made in May, 1900, the author used a new square cast-iron cylinder, similar to that shown in Fig. 3, but with four sides and so arranged that it could be either heated by steam, cooled by water, or kept at a constant temperature, and

the thermal effects thus studied. The temperature of the wall was taken in each case. When circulating water was used in the jacket, the quantity was weighed, and the rise in temperature noted. Indicator diagrams were also taken in each test from the low-pressure cylinder.

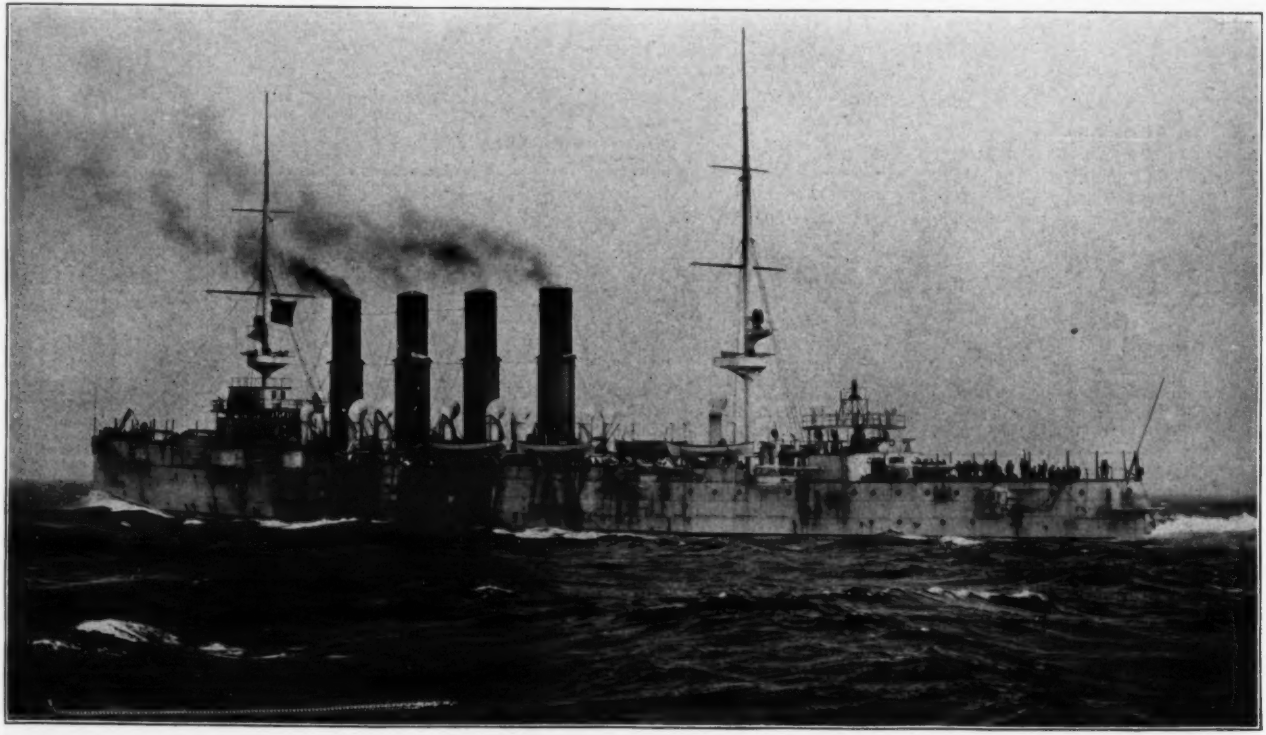
Six experiments were made with different pressures of steam in the jacket of the revealer cylinder, and, therefore, at different cylinder temperatures. At the start, and for some time after, the steam heated the cylinder to a given temperature. It was then shut off, and the effects noted every minute until a constant and minimum temperature was reached, due to the steam from the low-pressure cylinder. The object was to determine the different rates of cooling of the previously heated revealer cylinder. The only condition varied was the temperature of the little cylinder. The tests prove that, notwithstanding the water on both sides of the cast-iron surfaces, their heat-absorbing properties are greater than their heat emission to the exhaust.

Two other experiments were made. At starting the revealer cylinder was cooled by circulating water to a certain constant temperature. The water was then shut off and the rise in temperature of the cylinder noted every minute. The object was to determine the different rates of heating the previously cooled cylinder, and also whether the exchanges of heat were affected by more or less water on the cylinder surfaces due to condensation. The temperature of the cylinder only was varied by altering the quantity of circulating water. This seems to prove that this water does not interfere with the passage of heat through the cast-iron walls, and that the heat absorption goes on notwithstanding it.

Two experiments were made without shutting off the circulating water. The cylinder was cooled down to the desired temperature, which was then kept constant. One object was to determine whether the heat exchanged would be influenced by more or less condensing water on the external surfaces of the cylinder. Comparing these two tests, about twice as much heat passed in No. 2 from the low-pressure cylinder to the circulating water per unit of surface exposed. It had also to pass first

through a greater thickness of water or drops on the surfaces This seems to prove that this surface water does not interfere with the passage of heat through the cast-iron cylinder walls and that the heat absorption proceeds all the same. Several of these experiments agree with M. Nadal's theoretical investigations, as published in the "Revue de Mécanique," with the conclusions he draws—namely, that the heat absorbing properties of cast iron are much greater than their heat emitting properties, and the former are little influenced by the drops or condensation on the steam-engine cylinder walls. The author has not yet made experiments with surfaces slightly oiled; all the above were carried out with fairly clean surface. A little oil from the steam cylinder might perhas have been on them in places, but they appeared quite dry and even a little rusty to the touch.

Experiments showed that the loss of heat by radiation of cast-iron cylinder at the rather high temperature of 298 degrees Fahrenheit includes the heat transmitted along the pipe to the cylinder cover.



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IMPERIAL RUSSIAN CRUISER "VARIAG."

SPEED TRIALS OF THE *VARIAG*.

The Russian twin-screw, protected cruiser *Variag* built for the Imperial Russian Government by the Wm. Cramp & Sons' Ship and Engine Building Company, of Philadelphia, Pa., and concerning the phenomenal speed of which so many press comments have been made during the past four months, accomplished her contract performance satisfactorily on October 4, 1900, maintaining a speed of 23¼ knots for twelve consecutive hours (9 A. M. to 9 P. M.), and exceeding the requirements by ¼ knot. This was under natural draft, and breaks the records both for speed and for duration of test for full-speed natural-draft trials.

This speed was deduced from the standardization data of the propeller obtained by previous runs over a ten-mile course in July last, four runs (two each way) being made over the course at each of four different speeds, far enough apart to establish points for plotting a speed revolution curve, and from which the speed of the final trial could be obtained simply from the revolutions. The results of these standardization runs were:

No. 1.—Speed, 16.122 knots; average revolutions, main engines, 100.2; I.H.P., main engine, 4,371.

No. 2.—Speed, 19.6 knots; average revolutions, main engines, 119.7; I.H.P., main engines, 6,975.

No. 3.—Speed, 21.97 knots; average revolutions, main engines, 140.8; I.H.P., main engines, 12,107.

No. 4.—Speed, 23.7; average revolutions, main engines, 153.4; I.H.P., main engines, 16,206.

An attempt was made on July 28, 1900, to run off the contract trial, but after about 7½ hours run, during which an average speed of nearly 23.6 knots was secured, an accident to the port H.P. cylinder brought the test to a close for the time being, and the ship returned to the builders' works for repairs.

After repairs had been made, the official run was successfully made, as noted in the first paragraph of this article.

From the reports of the contractors we are assured that the final 12-hour trial was made without any "hitches" of any kind, and without any occurrences to cause the slightest uneasiness in regard to the engines, boilers or ship. No attempt was made to do the best possible, but only to maintain the revolutions necessary to safely secure the required results, so that the contractors certainly deserve congratulations and praise in reaching the objective in this cruiser.

The complete data of this last run has not yet been worked out, as the immediate desideratum was merely to be assured that the average revolutions of the engines was not below the 23-knot point. We are, however, able to give the principal data of the higher performance, that of the 7½ hours' trial on July 28, and this is tabulated below, after the ship and engine data.

While it is noted that natural draft only was used, the 60-inch fire-room blowers were kept constantly running, except when some were temporarily shut down, in order to hoist ashes. This is what might be termed assisted draft, although there was no air pressure in the fire room, by gage.

The *Variag* is equipped with a great number of electric motors, the after capstan, turbines for drainage, hull and engine-room ventilating blowers, forced-draft fans, ash hoists, winches and ammunition hoists are all so driven.

HULL DATA.

Length between perpendiculars, feet	400
on L.W.L., feet and inches	419–5
over all, feet	425
Beam, feet	52
Draught, feet and inches	19–6¾
Displacement, about, tons	6,500
Tons per inch of immersion	35.84
Foot tons to change trim one inch	726.10
Radius of action with normal coal at full speed, about, knots	1,000
10 knots speed, about, knots	3,000
Coal when all bunkers full, tons	1,290

Radius of action when all bunkers full at full speed, about, knots	1,700
10 knots speed, about, knots	5,000
Coal, normal, tons	770

The hull is constructed with double bottom but not sheathed.

There is a protective deck running the entire length of the ship, 1½ inches on the flat and 3 inches thick on the slopes.

Accommodation will be made for 21 officers, 9 petty officers and 550 crew. Provision for 60 days, and water for 10 days.

The armament will be as follows:

12 6-inch guns, caliber 45.
12 75-mm. guns, caliber 50.
6 47-mm. guns, Hotchkiss, one barrel.
2 Baranovsky's guns.
2 37-mm. Hotchkiss guns.

The ship will be furnished with the following boats:

2 40-foot steam launches.
2 12-oar cutters.
1 16-oar barge.
1 14-oar working cutter.
2 6-oar whale boats.
2 6-oar yawls.
Total, 10 boats.

Provision has been made for excellent supply of make-up feed water; 600 tons can be carried in the double bottoms if desired, and 60 tons will be regularly so carried in the fire-room double bottoms. Eight evaporators of 4,500 gallons each are installed, four of which are to be used only for make-up feed, with attachments, if necessary to connect all eight evaporators for this purpose.

ENGINE DATA.

Two 4-cylinder, inverted, vertical 40-62-(2) 68 by 36-inch stroke.	
Diameter crank shaft, 7-inch hole, inches	16½
line shaft, 7-inch hole, inches	15½
screw shaft, 7-inch hole, inches	16½
Cooling surface in condensers, square feet	24,000
Air pumps: Two independent, vertical D. S., 32 by 18 inches.	

Circulating pumps, impeller, inches	54
All cylinders, jacketed.	
All valves, piston.	
Cut-off, all engines, per cent	75
Propellers, blades	3
diameter, feet and inches	14–6
pitch, feet and inches	18–3
Feed pumps, ten	six 12 × 7 × 12 and four 9 × 5 × 12.
Fire and bilge pumps, two	10 × 7 × 12

BOILERS.

Description	Niclausse.
Number	30
Furnaces	30
doors	90
Number elements	420
tubes in each element	22
Total number tubes	9,240
Length of tubes, feet and inches	7–4
grate, feet and inches	6–5½
Grate surface, square feet	1,575
Total heating surface, including headers, square feet	62,229
H.S. ÷ G.S.	39.6
Outside diameter tubes, inches	3½
Number of chimneys	4
Height above grate, feet	90
Volume water in boilers (total) at steaming level, cubic feet	3,931
in steam room, cubic feet	875
Working pressure, pounds	256

RESULTS OF OFFICIAL TRIAL.

2·03 P. M. to 9·30 P. M., July 28th, 1900.

	Port.		*Starboard.*	
	M.E.P.	I.H.P.	M.E.P.	I.H.P.
Average H.P. cylinder	84.5	2,965	85.9	3,018
M.P. cylinder	27.6	2,327	28.2	2,373
1st L.P. cylinder	11.3	1,143	12.1	1,232
2d L.P. cylinder	11.8	1,195	12.1	1,233
	M.R.P.		M.R.P.	
one engine	37.82	7,631	38.97	7,864
two engines		15,495		
revolutions	152.9		152.75	
opening of throttle	.15		.15.	
steam	232.3		230	
1st receiver	191.3		190	
2d receiver	51.2		56.1	

Average 3d receiver	9.6	10.5
vacuum	23.3	22.5
cut-off H.P., inches	4	3¼
M.P	.740	.740
1st L.P	.753	.753
2d L.P	.709	.709
Steam to receiver	Closed.	Closed.
Average D.S. air pump	13	14.7
revolution circulating pump	190	173
temperature injection, degrees Fahrenheit	62	62
discharge, degrees Fahrenheit	122	126
hot well, degrees Fahrenheit	86	82.5
starting platform, degrees Fahr.	97	96
Mean speed, knots	23.6	
slip, per cent	15.4	
draught, feet and inches	19–6¾	
Displacement, tons	6,500	

FIRE-ROOM CONDITIONS.

2 P. M. to 9·25 P. M., July 28, 1900. Average pressure on all thirty boilers, 243.5 pounds.

Fires thin, about 9 inches thick. Firing good.

One-half glass of water carried on all boilers.

Outside of uptake hot, between 300 and 400 degrees Fahrenheit.

Temperature of passage between boiler and bulkhead, 127 degrees Fahrenheit.

No. 1. Forward Single Fire Room.

Average speed feed pump, double strokes per minute	29¼
pressure at pump, pounds	289⅝
steam pressure, pounds	249
temperature of feed water at pump, degrees Fahrenheit	81½
number revolutions of blowers	402⅝
air pressure	0
temperature of fire room on floor, degrees Fahrenheit	90⅜
gratings, degrees Fahrenheit	114¼

No. 2. Forward Middle Fire Room.

Average speed feed pump, double strokes per minute	26
pressure at pump, pounds	357
steam pressure, pounds	243⅝
temperature of feed water at pump, degrees Fahrenheit	83½
number revolutions of blowers	400⅞

Average air pressure..... 0
temperature of fire room on floor, degrees Fahrenheit..... 92½
gratings, degrees Fahrenheit..... 112½

No. 3. Middle Fire Room.

Average speed feed pump, double strokes per minute..... 23⅞
pressure at pump, pounds..... 378½
steam pressure, pounds..... 248 1/64
temperature of feed water at pump..... Thermometer broken.
number revolutions of blowers..... 396⅜
air pressure..... 0
temperature of fire room on floor, degrees Fahrenheit..... 98⅜
gratings, degrees Fahrenheit..... 123

No. 4. After Middle Fire Room.

Average speed feed pump, double strokes per minute..... 36⅛
pressure at pump, pounds..... 410⅜
steam pressure, pounds..... 238¾
temperature of feed water at pump, degrees Fahrenheit..... 84 1/14
number revolutions of blowers..... 398½
air pressure..... 0
temperature of fire room on floor, degrees Fahrenheit..... 94⅜
gratings, degrees Fahrenheit..... 134⅞

No. 5. After Single Fire Room.

Average speed feed pump, double strokes per minute..... 28⅝
pressure at pump, pounds..... 345⅝
steam pressure, pounds..... 240 1/14
temperature of feed water at pump, degrees Fahrenheit..... 84½
number revolutions of blowers..... 387⅜
air pressure..... 0
temperature of fire room on floor, degrees Fahrenheit..... 103⅞
gratings, degrees Fahrenheit..... 157

BLOWER CONDITIONS.

The motors ran well; no sparking; bearings cool. Foundations firm; no vibration.

No. 1 blower shut down from 7·10 P. M. to 7·55 P. M. to permit hoisting ashes.

No. 3 and No. 4 blowers shut down 7·20 P. M. for same reason.

No. 9 and No. 10 blowers shut down 7·35 P. M. for same reason. Were still shut down when trial ended 9·25 P. M.

Boiler held steam well after blowers were stopped.

COMPARISON OF PROTECTED CRUISERS.—TABLE No. I.

	United States built.			English built.			
	Russian *Variag.*	U. S. S. *Minneapolis.*	U. S. S. *Olympia.*	*Terrible.*	*Diadem.*	*Blanco Encalada.*	*25 De Mayo.*
Length on load line, feet and inches	419-6	412-0	340-0	520-0	435-0	390-0	345-0
Beam, extreme, feet and inches	52-0	58-2½	53-0	71-6	69-0	45-9	43-0
Trial draught, feet and inches	19-6	22-6½	21-6	27-0	25-3	18-5	16-0
Trial displacement, tons	6,465	7,375 *	5,586 *	14,200	11,000	4,568	3,200
Indicated horsepower	15,925	20,862	17,313	25,000	16,961	14,500	13,800
Speed, knots	23.25	23.07	21.69	22	20.72	22.80	22.43
Duration of trial	12 hours.	4 hours.	4 hours.	.....	8 hours.	Measured mile.	Measured mile.
Coal capacity, tons	1,250	1,891	1,170	3,000	.....	866	.....
Coal on trial, tons	770	629	153	1,000	1,000	350	300
Battery	12 6″ R.F. 12 75-mm R.F. 6 47-mm. 2 37-mm. 2 field.	1 8″ B.L.R. 2 6″ R.F. 8 4″ R.F. 12 6-pdr. R.F. 4 1-pdr. 4 Gatlings.	4 8″ B.L.R. 10 5″ R.F. 14 6-pdr. R.F. 6 1-pdr. R.F. 4 Gatlings.	2 9.2″ 12 6″ R.F. 18 12-pdr. 12 3-pdr.	16 6″ R.F. 14 12-pdr. R.F. 12 3-pdr. R F	2 8″ R.F. 10 6″ R.F. 12 3-pdr. 10 1-pdr.	2 8″ R.F. 8 4.7″ R.F. 12 3-pdrs. 12 1-pdr.
Complement	580	477	446	840	600	429	285
Type of boilers	Niclausse.	Scotch.	Scotch.	Belleville.	Belleville.	Scotch.	Scotch.
Type of engine	4-cylinder triple expansion.	Triple expansion.	Triple expansion.	Triple expansion.	Triple expansion.	Triple expansion.	Triple expansion.
Number of screws	2	3	2	2	2	2	2
Fixed weights or weight of hull, boilers and engines, in tons	4,706	6,096	4,696	10,702	8,412	3,568	2,516
Weight of ordnance in tons	547	230	327	.....	652	400	210
Weight of equipment and outfit in tons	442	420	400	.....	936	250	140
Weight of coal in tons	770	629	153	1,000	1,000	350	300
Total weight of ordnance, coal, equipment and outfit in tons	1,759	1,279	880	3,498	2,588	1,000	650
Percentage of weight of ordnance, coal, equipment and outfit to weight of hull, boilers and engines	37.40	20.90	18.70	32.60	23.50	28.20	25.90
Percentage of coal to trial displacement	12	8.40	2 70	7.00	9.10	7.70	9.60
Percentage of ordnance to trial displacement	8.40	3.10	5.70	.....	5.90	8.80	6.60
Speed	23.25 knots for 12 hours, natural draft.	23.07 knots for 4 hours, forced draft.	21.69 knots for 4 hours, forced draft.	22 knots ? for 8 hours, ? natural draft.	20.72 knots for 8 hours, natural draft.	22.80 knots, measured mile, forced draft.	22.43 knots. measured mile, forced draft.

* Data taken from report of Trial Board.

AVERAGE NUMBER OF REVOLUTIONS.

No. 1	blower,	$406\frac{2}{7}$
No. 2	"	$399\frac{1}{8}$
No. 3	"	408
No. 4	"	$402\frac{1}{5}$
No. 5	"	$399\frac{1}{2}$
No. 6	"	$394\frac{7}{8}$
No. 7	"	$397\frac{1}{7}$
No. 8	"	$393\frac{3}{7}$
No. 9	"	392
No. 10	"	$402\frac{4}{5}$
No. 11	"	$405\frac{3}{8}$
No. 12	"	399
No. 13	"	$393\frac{5}{8}$
No. 14	"	$396\frac{3}{4}$
No. 15	"	$387\frac{1}{8}$
No. 16	"	$387\frac{5}{8}$

Combined average number of revolutions for sixteen blowers, $397\frac{7}{16}$.

A thorough examination of the boilers after this run showed no signs of distress.

Mr. Chas. H. Cramp, in a paper read before the Society of Naval Architects and Marine Engineers at the last meeting, submitted two tables of comparisons between the *Variag* and several other noted protected cruisers of English and American build, which we have here reprinted.

TABLE No. II.

—	Trial-trip displacement.	Fixed weights: hull, boilers and engines.	Total weight of ordnance, coal, equipment and outfit.	Percentage of weight of ordnance, coal equipment and outfit to fixed weights: or weight of hull, boilers and engines.
Variag	6,465	4,706	1,759	37 7
Minneapolis	7,375	6,096	1,279	20.9
Olympia	5,586	4,696	880	18.7
Blanco Encalada	4,568	3,568	1,000	28.2
25 De Mayo	3,166	2,516	650	25 9
Diadem	11,000	8,412	2,588	23.5
Terrible	14,200	10,702	3,498	32.6

NOTES.

ANNUAL REPORT OF THE CHIEF OF THE BUREAU OF STEAM ENGINEERING.

The Report of the Engineer-in-Chief is somewhat more voluminous this year than last, and, although submitted to the Department on October 1st, it has been delayed in issue in pamphlet form by the tedious work of getting the illustrations reproduced; these illustrations being of the newly-designed machinery for battleships and cruisers and also of other drawings referred to in the Report. The text, however, was given to the press by the Department and has received numerous appreciative comments upon its most important features.

We give here the index of the several *heads* under which the report is made, to mark its scope, and have also quoted a few remarks from the "American Machinist," whose editor's "brief" is not too long to reprint here.

The headings are as follows: Expenditures for 1899–1900—Designing work in Bureau—Repair work—Repair ship—Vessels under special repair—Inspection work, general—Consolidation of bureaus—Personnel—Naval Academy—Warrant machinists—Drafting room, details of work in—Material inspection, details of—Machinery plants, improvements in—Laying up ships—Torpedo boats, care of—Experimental work—Liquid fuel—Boilers for *Cincinnati* (Babcock & Wilcox water tube), test of one—Navy yards and stations—Trial trips of *Kentucky*, *Alabama*, *Albany*, *Isla de Luzon*, *Isla de Cuba*—Vessels under construction—Machinery of United States naval vessels, data of—Estimates of expenditures for 1901–1902.

In commenting upon this paper the "American Machinist" says:

The annual report of Admiral George W. Melville, Chief of the Bureau of Steam Engineering, recently forwarded to the Secretary of the Navy, is, as it could not have failed to be, an ably written and highly interesting paper. We can spare space only to notice a few of the most important points touched upon.

The paper opens with a plea for appropriations adequate for maintaining the work of the Bureau, so that it may be most efficient, and not incomparable with the other nations of the world, pointing out the activities of all the powers and the lavish expenditures not only in repairs and maintenance, but also in the building of new ships, and especially in conducting elaborate and extensive experimental work in the various fields of investigation. The necessity of building and maintaining repair ships is especially insisted upon, the work of the hastily improvised *Vulcan* in the Spanish War having amply demonstrated the value of such adjuncts to fleets engaged in the actual operations of war. An appropriation is asked for a new ship throughout of at least 5,000 tons displacement and to cost about $1,000,000, which, by the way, would only be about one-quarter of the cost of a first-class battleship.

The improvements under way at the shops and machinery plants at the various navy yards come naturally next in the report. There has been delay in placing contracts for new buildings on account of the high price of material, and changes have had to be made in designs to keep within the appropriations. Great care has been taken to make all new buildings absolutely fireproof. Contracts have been made for the power plant of the new shop at the New York navy yard, and also for the power tools.

In the preparation of specifications for tools, care was exercised to avoid calling for patented and proprietary articles which could be supplied only by individual firms, and the success of the scheme was evidenced by the great numbers and the wide range of the bids received, and a selection of first-class tools was made at reasonable prices. An extensive and fully-equipped plant is called for in the Philippines, either at Cavité or at some more suitable point.

An appropriation of $100,000 is asked for experimental work. Immense sums are expended annally by other powers, and the results obtained lead to the avoidance of costly errors in practice, and to higher efficiencies in succeeding designs. The unsatisfactoriness of experiments made with models is sufficiently

familiar. The Bureau needs reliable data on the economy of engines running at a small fraction of their maximum power, and this information is to be obtained only from a modern and fully equipped vessel. The economy of the boilers of the main engines and the various auxiliaries could be obtained for various speeds, and the data thus obtained would suggest changes for the better in all subsequent practice. The *Raleigh* or the *Cincinnati* is suggested as a suitable vessel, which could at the same time be a practice vessel for the Naval Academy.

The use of liquid fuel has not been demonstrated to be more desirable than that of coal, either here or abroad. Suggestions are made towards the imposing upon contractors for new ships of requirements for economy tests whereby the machinery at maximum and other selected powers must show certain horsepower at stipulated expenditure of coal. This is insisted upon by other powers, but, with us, as detailed plans of all machinery are examined and approved by the Department, such requirements have seemed to be scarcely just.

The report strongly opposes the consolidation of the Bureaus of Construction and Repair, Steam Engineering and Equipment into one Bureau of Ships, as proposed by the Secretary of the Navy a year ago. "Every tendency is to increase specialization in work, owing to the ascertained value of a system whereby leading talent in each department may devote itself to the advancement of its particular branch, and not be diverted therefrom through a maze of miscellaneous and unrelated work." The next paragraph which we reproduce is full of the personality of the author.

"That differences should agitate the board (referring to the present Board on Construction) is not only natural, but also highly proper, and is a most effective protection against hasty and regrettable action. This occasional lack of harmony must not be confused with any real lack of efficiency, nor into a cause for recommendations having for their object the reversal of the true business principles by a combination of radically different professional duties. Not only would such a course fail to bring harmony, but a surer result would be in a speedy decline

in the rate of progress made in each profession. I, in common with the chiefs of other bureaus, naturally feel a pride in the eminence of the Bureau I control, but I am honest in declaring that my opposition to the proposed combination has a reason far above personal feeling or ambition, and that reason is my positive knowledge of the fallacy of the arguments for it and the firm conviction that only confusion, greatly decreased efficiency and greatly increased expense can possibly result."

A most regrettable thing is the necessity which exists for the report to go again into a discussion of the personnel question. The effect of the law after two years of trial, the report states, has been very unsatisfactory to the steam engineering interest of the Navy. The work of the Bureau has greatly increased while the number of skilled officers available for the work has decidedly decreased. The cause lies not so much in the law itself as in the failure of the Department to encourage officers to take up this line of work. The concluding portion of the report is a most earnest appeal for efficient engineer officers in sufficient numbers, and most serious consequences are predicted if prompt steps are not taken to materially increase the present staff. Admiral Melville says:

"In attention to my recommendations or apathy regarding the immediateness of the necessity for more active and decided measures toward securing the desired conditions can surely result in nothing but rapidly decreasing efficiency, from which it will be continually more difficult to recover, and the cost of which will be significant in enormously larger repair bills, shorter-lived machinery, and a fleet of vessels in doubtful fitness for their designed service—a Cervera fleet, with limitless men, but lacking the technical experts needed to meet the extraordinary and ever new conditions of emergency and war.

"I regret I have failed to impress you to the point of action by my former communications. Had a series of calamitous events occurred during the past year to make graphic the insufficiency of the present force of expert engineers, I am sure potent remedial measures would have been promptly taken by the Department. But while glad indeed to have disasters

averted, I can assure you that danger now exists. It lurks in the silence of seeming security, but a knowledge of its presence should increase the desire to hasten its removal. Fortune alone has postponed casualty.

"The country can safely count on the valor and fidelity of its officers and men, but fidelity and valor without knowledge of the use of the arms given them with which to do battle can avail little against an efficiently drilled foe, and will scarcely afford more than an exhibition of heroic sacrifice, as needless as it would be cruel. The arms of a battleship are her machinery and her guns, 'useless each without the other,' and strong to victory when working well together. No deep thought is necessary to understand this, in the light of late experience. The very highest degree of excellence in both the condition and handling of each is the price of successful encounter, or, at least, is the expectation of the country. A ship, motionless or helpless to maneuver well, could never make efficient battle, be her guns ever so good or her crew ever so brave. To guard against this the head of the Steam Engineering Department must be full of resource and armed by experience and engineering ability, only attained by years of intimate association with machinery under all conditions of service. Hap-hazard luck may bring a ship through without this, but sane judgment would condemn dependence on simple fortune or a failure to use every possible effort to insure a most competent management in this most important of ships' departments."

The Admiral would, by departmental order, compel all line officers below the navigators of ships to alternate in duty in the engine room and on deck, and would have efficiency reports made quarterly to note their progress and to class their ability.

BRITISH PRIZE NAVAL ESSAY.

("ARMY AND NAVY JOURNAL.")

If the editors of the daily newspapers that made such a fuss over Lieut. Hobson's remark that the Spanish ships at Manila were sunk by their crews and not by Dewey's guns had read the

naval prize essay published in the "Journal of the Royal United Service Institution," they would have known just how largely gun fire has figured and what part it has played in the destruction of fleets in the wars of the last decade. The luminous details that have been published of the Manila and Santiago sea fights ought to have left little ignorance among those who are supposed to follow current history as to the real cause of the destruction of the Spanish ships at Manila. Lieut. Hobson was sent to Santiago to raise a Spanish warship. Suppose he had returned and said that the ships had not been sunk by shell fire, but by being run on the rocks, would any one have construed the remark as reflecting on the efficiency of the fleet of which Hobson himself was a part? The *Cristobal Colon* was run ashore when Capt. Moreu found he would be cut off by the *Brooklyn*, and when a big shell from the *Oregon* struck near him. So far as injuries were concerned, the *Colon* was almost as sound when she ran ashore as when she issued from the harbor. Yet she was as much a victim of our unerring pursuit as if she had been sunk miles from shore. In fact all of Cevera's four large ships were run ashore, thus being directly wrecked by their own crews, but this fact was never used to belittle the work done by our ships.

Among the officers and men of our navy the misrepresentation of Lieut. Hobson's remark was so clumsy that it carried its own correction on its face and scarcely needed the explanation by the Lieutenant, on his arrival at the New York Navy Yard last week, that he had no intention of depreciating the work of Admiral Dewey's ships, and merely stated the fact that the Spanish sailors had opened the seacocks of their ships, preferring to sink them than to have them fall into the enemy's hands.

The prize essay to which we have referred was written by Major O. Field, R. M. L. T. It discusses the best type of war vessels for the British Navy in the light of the changes in naval construction during the past twenty years, and of the experience gained in the Chino-Japanese and Spanish-American wars. The treatment of the question includes armor, equipment and armament. In these two wars, he says, "the causes which led to the destruction of the many Chinese and Spanish ships at the Yulu,

at Manila and at Santiago were in almost every case the conflagrations that broke out on board and the demoralization of their crews by these and by the torrent of projectiles that riddled their unprotected batteries.

"At the battle of Santiago the *Infanta Teresa*, the *Viscaya* and the *Oquendo* were all put out of action by the effect of the American fire on their hulls above the water line, which wrecked everything, made a shambles of their batteries, and set their decks and woodwork ablaze, while the *Colon* was run ashore by her commander when escape was impossible. In no case was there any damage at the water line sufficient to imperil the flotation of the ship."

To the essayist it appears "as if we (the British) had, comparatively speaking, approached finality. Not only do we concontinue to build battleships so much alike that a squadron of them presents the same uniform appearance that the fleets of Nelson, with their long lines of checker-sided three-deckers do in the pictures with which we are so familiar, but foreign nations are copying the type more or less closely in their newest naval creations. The Russians, the Germans, the Italians and our cousins across the Atlantic all now build their battleships with their principal armament of four heavy guns disposed in pairs in turrets or barbettes, fore and aft on the central line of the ship, with a central battery of heavy Q.F. guns between them. Even the French with their great shipbuilding prestige seem now inclined to abandon their favorite types to approach more nearly to our own."

Eight lessons are found by Major Field in the two wars. These are: the urgent necessity for the removal of every possible portion of woodwork which now enters into the interior construction of a man-of-war, and as a corrollary the advisability of improving the protection of the fire mains; the necessity of providing thorough protection for the guns' crews; the fact that armor plating has shown itself more effective than had generally been supposed; the uselessness, not to say danger, of lightly armored gun shields; the suicidal nature of the custom of mounting torpedo tubes above water; the comparatively slight effect

of the heaviest guns when their cost, weight and slowness of fire are considered; the very great value and efficiency of Q.F. cannon, especially the heavier classes of weapon; the importance of homogeneity in a fleet or squadron.

The "negative inferences" he draws are these: The absence of any deliberate attempt to make use of the ram, and the comparatively little effort to utilize the Whitehead automobile torpedo as a weapon of offence.

The effect of the woodwork on the Spanish ships at Manila is not discussed by the author, as he says their destruction was inevitable, since they were feeble and out of date. The damage done by fire on the *Viscaya*, *Oquendo* and *Maria Teresa* at Santiago was so appalling that the Americans promptly ordered the removal of all woodwork on the *Chicago* and *Newark*, in process of reconstruction.

The essayist does not agree with the contention that if we make ships more uninflammable we can afford to leave a large part of them open to the free passage of explosive projectiles, as the great penetration of fragments of shell exposes to great loss of life, not to speak of the dangers of asphyxiation from the fumes of new explosives. As to greater effectiveness of armor-plating, the most valuable illustration is found in the case of the *Colon* at Santiago, from which a 5-inch armor-piercing projectile, striking at an angle of 45 degrees, rebounded, though capable theoretically of piercing twelve or thirteen inches of iron, the *Colon's* armor being only 6-inch Harveyized. The fact that the *Colon* was the only Spanish ship to hit the *Brooklyn* with anything heavier than a 6-pounder suggests that her extensive armor protection gave greater confidence to her gunners.

Need of Thick Gun Shields.—The danger of thin gun shields had a striking illustration at Santiago when the hood of the foremost 11-inch gun was penetrated by an 8-inch shell, which killed everyone in the barbette. The *Maria Teresa* also suffered in a similar way, and the essayist's conclusion is that shields, unless at least four or five inches in thickness, are more harmful than useful. There is nothing for it but to substitute regular armored protection for every gun in the secondary batteries.

As to the peril attending the use of above-water torpedo tubes, the paper says the Spanish-American war proved the truth of the predictions that above-water torpedo tubes would prove disastrous. At Santiago the *Oquendo* is supposed to have had a torpedo exploded by a 5 or 6-inch shell with disastrous effect; two 12-inch projectiles, which burst in the after torpedo room of the *Infanta Maria Teresa*, are thought to have caused the explosion of others, while there was after the fight a gaping hole in the starboard bow of the *Viscaya* which was also put down to a similar cause.

Coming to the effect of the heavy guns, Major Field points out that when the Spanish ships were examined at Santiago by a board of experts no hit by any 13-inch projectile could be discovered, and only two made by 12-inch shell. Many of the shell from the 8-inch guns carried by the American ships would seem to have done almost as much damage, and as the eighteen of them which were in action made between them a dozen hits there is no doubt that they proved much more formidable to the enemy than did the fourteen very heavy guns with their poor score of two hits.

The great value and efficiency of Q. F. cannon was amply demonstrated at the battle of the Yalu by the Japanese, who had batteries of 4.7-inch quich-firers in nearly all their ships which were present. The Chinese, on the other hand, are not supposed to have had more than two or three of these guns in their whole squadron, and no quick-firers of higher caliber. They had a considérable number of 3, 4 and 6-pounder rapid-firers, but so also had their opponents. But with the two or three Q. F. 4.7 guns that they had and a couple of breech-loaders of the same caliber the Chinese contrived to hit the *Sakio* eleven times. In fact, this scoring compares favorably with the record of the American 4 and 5-inch quick-firers at Santiago. Of the two calibers together they had nine in action, and made between them twenty-seven hits, an average of three a gun. The Chinamen, too, were under a much more severe fire than the Spaniards were able to direct upon the American gunners. But in both battles the greater part of the havoc and destruction may be safely credited to the

quick-firers. Their closest rival without doubt has been the 8-inch slow-firing breech-loader, which was extensively used by the Americans both at Manila and at Santiago. This probably was on account of the greater size of its projectiles and of their bursting charges, combined with a fair rapidity of fire. The paper favors substituting 8-inch quick-firers for some, at any rate, of the 6-inch weapons now carried on British ships.

The importance of having an ample supply of ammunition for Q. F. guns and a means for its rapid transference is brought home by recent experiences. At the Yalu, disaster fell upon the *Matsushima* from the explosion of a pile of cartridges which had been accumulated in an unprotected position. At Manila, after two and a half hours' fighting, Admiral Dewey drew his ships out of range to take stock of his ammunition. A ship armed with Q. F. cannon must have a rapid means of supplying them with a constant stream of projectiles from below the water-line by means of hoists continuously running and actuated by small electric motors, or other means if considered preferable.

The importance of homogeneity was shown at the Yalu, where the Chinese might have made a different fight had they had a couple more ironclads like the *Chen Yuen* and *Ting Yuen*, in the place of the numerous but heterogeneous collection of cruisers and gunboats. Want of homogeneity, too, considerably reduced the effectiveness of the Japanese fleet with which they were engaged. The importance of homogeneity was not conspicuously demonstrated by any of the events of the Spanish-American war. The Americans had the whole affair their own way from start to finish, and though their fleets which were engaged in the Manila and Cuban campaigns were not particularly homogeneous, yet they far and away outmatched the Spaniards, as, indeed, they would have done had every ship at their disposal been of different speed, tonnage, armor and armament.

As to the use of the ram, the paper says that at Santiago the Spaniards might, perhaps, have employed it as a last resource with some effect. But they were too fully occupied in trying to escape to attempt it. The *Vizcaya's* movement toward the

Brooklyn, Major Field thinks, was not to ram, but to use the bow torpedo.

On the use of the automobile torpedo, the essayist says that no lesson or theory can be drawn from the failure of either the Chinese or Spaniards to use their torpedo boats to attack the blockading squadron off Wei-hei-Wei or Santiago, for initiative was absolutely wanting in the officers of both nations. The only remark that can be made in reference to this is, that to make effective use of torpedo boats a bold and efficient body of officers and men is a *sine qua non*. These are most likely to be found in the finest navies, and not in the second-rate ones. The Americans, very possibly, might have made a successful torpedo attack on Admiral Cervera's squadron as it lay at anchor in the harbor of Santiago; but, having the Spanish ships well bottled up, they might reasonably hope that they might fall into their hands intact sooner or later. "Still, had they made an early and successful attack, the work of the army would probably have been much shortened and the defense of the city would have most likely collapsed much sooner than it did."

Major Field thinks that if the Spanish destroyers *Pluton* and *Furor* had kept on the off-side of their cruisers it is just possible that they might have got a chance of using their torpedoes with some effect. "They were handled in a way that can only be described as a compound of madness, fatuousness and ignorance, and, as might have been foreseen, were literally blown out of the water in a very few minutes. To suggest as a lesson that this procedure is inadvisable, would be to insult the intelligence of the officers of any navy in the world."

The Chino-Japanese and the Spanish-American wars were both too one-sided, and especially so in the initiative, to test the value of the automobile torpedo; but too much must not be expected of the weapon, even against a comparatively inactive enemy.

Concerning the future warship type the paper considers "that our latest types approach very nearly to the ideal. No men-of-war belonging to foreign powers carry such ample supplies of coal, stores and ammunition as do our own (British); in no navy

in the world is there the same homogeneity in the various classes of war vessels. The advantage of this last factor can hardly be overestimated, though, strange to say, it seems only lately to have been grasped by our rivals abroad."

Major Field comes out squarely for big battleships of not less than 12,000 tons displacement. The cost of maintenance and upkeep of such a ship is on her side when comparison is made with smaller ships. The *Canopus* and *Royal Sovereign* type furnishes a design "that is an excellent one," the only points open to criticism being the amount of protection afforded to the 6-inch guns, their comparatively small number and the unarmored ends. The paper then discusses with the aid of plates the different styles of vessels. The proposed commerce protector is to be of from 12,000 to 15,000 tons. In outward appearance it will be assimilated as closely as possible to a big mail boat, not only in order to give her a better chance of approaching an enemy's cruisers without arousing suspicion, but also to create among them a feeling of caution when coming up to a mail boat. It should have no military tops and the funnels should be only two in number. It should be able to stand the closest scrutiny from the enemy's field glasses. Dealing generally with cruisers, the paper admits the justice of the American criticism that the British cruisers are under-gunned.

No important divergence is suggested from the present types of torpedo boats and destroyers. The activity of France and the United States in pushing submarine experiments does not penetrate the British apathy in this direction, and Major Field dogmatically declares that "submarines are a confession of weakness and by no means to be recommended for our own navy, whatever foreigners may think of them."

Protection for the crews is the burden of the essayist's concluding remarks. The awful loss of life on the Spanish ships, as a result of the deluge of American projectiles, has evidently made a deep impression on the mind of the English naval experts, if the paper may be taken as a criterion. Added armor, he insists, should be provided even if the number of guns must be reduced, or even to a small extent the coal. While granting certain

advantages to come from the dispersion of gun positions, he asks whether the gain is enough to compensate for the danger incurred by leaving the most important portion of the main deck open to destruction, and whether it is not better to concentrate the guns in order that a complete armor defense may be provided for them, and, instead of endeavoring to avoid danger by dispersal, boldly defy attack by adequate protection. If the flotation of the ship is also secured, and the unarmored portion above the water line, fore and aft of the citadel rendered fire proof, its riddling and destruction would be a matter of comparatively little consequence. No one would be in it in action; no one would be called on to leave the guns and the armored citadel to put fires out in it, and it is possible that an enemy, recognizing the futility of expending his ammunition on them, would restrict his fire, as far as he could, to amidship, although here the ship would be best fitted to withstand it.—"Army and Navy Journal."

ALUMINUM-THERMIT.

The high reducing power of aluminum was recognized and studied by its discoverer, Wöhler, in 1827; by Ste. Claire Deville, who first produced the metal on a commercial scale in 1845; by Tessier, Troost, Mallet, Cl. Winkler, Green and Wahl, L. Franck, and by almost everybody who worked in the matallurgy of aluminum. All these investigators heated the crucible, which contained the mixture of aluminum and the substance to be reduced, and as the reaction is sudden and very energetic, almost explosive sometimes, *e. g.*, in the case of alkali sulphates, no practical process was devised for rendering the reducing action of aluminum available in the arts, and the crucibles never survived the operation. More recently, however, Dr. Hans Goldschmidt found that the mixture need not be heated from the outside; it is only necessary that the reaction be started, and this is done with the aid of a mixture of powdered aluminum and some peroxide, barium peroxide by preference. At the time he first described his process two years ago, he shaped the primer into an "ignition cherry," into which he stuck a piece of magnesium

wire. The Chemische Thermo-Industrie Gesellschaft, of Essen, which has since taken up the process, now supplies the primer as a white powder in metallic cases weighing from 2 pounds to 10 pounds. The Société d'Electro-Chimie, of Paris, which works under a license from the parent company, has introduced it under the very appropriate term "alumino-thermie."

The reduction of metals by Dr. Goldschmidt's process is exceedingly simple. For instance, metallic aluminum and chromium oxide, both finely powdered, are mixed in crucibles of a capacity of several hundredweights, made of firebrick. The mixture is set alight, and the operation is completed within half an hour. The metals obtained in this way are pure, especially free from carbon — an essential point — provided, of course, that the materials were pure. The regulus is found under a layer of molten alumina, the natural corundum which we know, in various degrees of purity, crystallized or amorphous, as ruby, sapphire, or emery. The crust is said to give an excellent emery, and, being free from iron and hydrate, it ought to do so. Chromium and manganese and their alloys, ferro-titanium, ferro-boron, etc., are now manufactured on a large scale in this manner, and remarkably fine metals and alloys are obtained. The reduction of vanadic acid has, however, failed. The supposed vanadium, submitted to Hittorf, turned out to be the suboxide Vd_2O.

For engineers there has been prepared a substance called "thermit," a mixture of iron oxide Fe_2O_3 and aluminum. Its chief use is for repairs of large pieces which cannot be heated in the ordinary way. Suppose we have to replace a broken tooth of a large wheel. The tooth is moulded *in situ* in fireclay, clay and coke, with or without the addition of sand; ample dimensions are allowed. Then the required weight of thermit—thermit R is recommended for castings—is melted in the special crucible supplied by the Essen Company; 1 pound of thermit yields ½ pound of iron. We start with a pound or two of thermit, strew some primer, about 1 per cent., of this mass, on it, and ignite it by throwing a common fusee match on the top, or dipping a red-hot iron wire into the mixture. The crucible should have been placed in a safe spot, and the man should wear smoked

spectacles to protect his eyes against the glare. The reaction starts suddenly. More thermit is strewn into the white-hot mass, not thrown, until the calculated weight has become molten; the temperature is estimated at 3,000 degrees centigrade. The liquid corundum is now poured off, and then the iron, which is, of course, easily distinguished from the alumina, poured into the mould. The piece is slowly cooled, and the tooth hammered and filed down to exact dimensions. The filling in of holes in faulty castings is simpler still.

Rail welding by "thermit" was first tried on the Essen-Steele electric tramway line a year ago, and no fractures have occurred, we are informed, during the experience of a winter and a summer. In South Germany and in Brunswick railway tracks have been treated in the same way, and this process appears to be regarded with favor. When we think of the heavy and costly machinery which has been employed in the United States for electric rail welding, we cannot help being struck with the ease of this new operation. We are aware that simple iron casting is also practised on rail joints. But so simple an outfit has hardly been seen before. Two men, a crucible, with a pair of tongs, a large case of thermit, and a small case of primer, and a box or mold, are all that is required. The mold is clamped to the rails. How long the joints will last remains to be seen. Welded joints have not exactly fulfilled the hopes their advocates placed upon them, we believe, though they have not justified the severe objections raised by their opponents either.

We come now to welding proper, in which the thermit is not used to furnish a cementing material, but only to act as a carrier of heat. Greater success appears to have been achieved in this direction than one might expect. The alumina is not poured away in this case, but surrounds the outer surfaces, so as to prevent the iron lump from fusing on to the work under treatment. Test bars welded with thermit gave a strength of 36.8 kilogrammes, against 35.8 kilogrammes per square millimeter before treatment, and an elongation of 11.5 per cent. against 15 per cent. previously. The tensile strength was thus impaired, but the fracture occurred some distance from the weld. Tubes,

which have been welded endways, have stood a pressure of 400 atmospheres, finally splitting lengthways when hammered or bent, but not breaking at the joints. This is a very noteworthy achievement, considering that only the edges of the tubes are in contact. Flanged tubes of this type would not have stood more than 200 atmospheres, it is asserted.

The welding operation requires a special two-part clamp to hold the tubes in straight alignment, and to support the mold for the liquid metal. This mold is either a ring-shaped casing encircling the joint or simply a box, open above, built up of sheet iron, which, for a pipe 2 inches in internal diameter, with a shell 0.16 inch thick, would have a length of 3 inches. Not to concentrate the heat below, the bottom of the box is brought a little nearer to the pipe than are the sides. The sheet metal of the box has a thickness of $\frac{1}{16}$ inch; outside it is packed with wet sand. The walls of the box need not make a good fit; a sheet or two of paper will prevent the metal from squirting out into the sand. When pouring the thermit in, care is taken that the jet does not directly fall upon the pipes to be joined, but rather hits the walls of the box. Thermit P is used in this case, and about 3 pounds of it will be required. The melting of this mass will not occupy more than two or three minutes, and the reaction proceeds so rapidly that the crucible hardly appears to get hot. The molten alumina, coming first in contact with the cold pipe, forms a crust on it which prevents union between the molten iron and the tube. The iron box is protected in the same way, so that the box can easily be removed afterwards. The operation will require some skill, we should think. Looking through the tube, the gradual heating of the inner wall can be watched; welding temperature would, in this instance, be reached within about $1\frac{1}{2}$ minutes, and the clamp bolts are then tightened. A few minutes afterwards the iron lump will have hardened sufficiently to be knocked off. This should be done without delay. The weld will not want any finishing.

The firm states that such a weld would, on average, cost from 2s. 6d to 4s., including cost of materials, crucible and wages; one man can see to small welds by himself. As to crucibles,

common Hessian or graphite crucibles will not do. The Essen Works have succeeded in preparing iron and clay crucibles, lined with magnesite and corundum, these crucibles are said to stand several castings. They must always be emptied completely, however, as any lump of iron remaining in the crucible would retard the fusion of fresh material. This is an additional reason why the quantity of thermit should be calculated and measured with a fair amount of accuracy. For 6-inch pipes (internal diameter), for instance, with a shell ¼ inch thick, 15 pounds or 18 pounds of thermit P are required. This thermit is a dark-brown powder, which is neither corrosive nor inflammable, and not more dangerous, under ordinary circumstances, than the coal which it replaces. The primer powder is, of course, inflammable, but it is indifferent to blows, as special experiments prove, and does not undergo spontaneous ignition. Both powders lose their efficacy when wet; the humidity of the atmosphere does not spoil them, however. Yet the boxes should be kept dry and in a safe place.—"Engineering."

THE NEW 750-FOOT DRYDOCK OF THE SAN FRANCISCO DRYDOCK CO., AT HUNTERS POINT, CAL.

A notable addition to the drydock facilities of the Pacific coast is planned by the San Francisco Drydock Co. at its yards at Hunters Point, San Francisco, Cal. The new dock for which plans have been prepared and the bids for construction asked will be 750 feet long, 122 feet wide at the coping, and 74 feet wide at the bottom, and will have a depth at the entrance of 32½ feet below the coping and 28 feet below high-water level. As will be observed from these dimensions, the dock will be one of the largest and most commodious in the world. Compared with the new masonry drydocks which are to be built by the United States Government the corresponding main dimensions are as follows:

Dimensions.	Government. ft.	ins.	California. ft.	ins.
Length of coping	750	...	750	...
Width on coping	144	6	122	...
Width on floor	80	...	74	....
Depth of floor below coping	39	3	35	10

The plans for the new Hunters Point dock have been prepared by Mr. Howard C. Holmes, Chief Engineer, State Harbor Commissioners, San Francisco, Cal., and we are indebted to this gentleman for the drawings and information from which the following description is prepared:

The new dock is located south of the present dock, with its axis veering at an angle of about 14 degrees from the axis of the old dock. Like the old dock, it will be excavated in the solid rock of the little peninsula which forms the property of the company. This rock is green serpentine, impervious to the water and easily excavated. This rock is to be excavated to lines corresponding closely to the contours of the dock chamber so that only a thin lining of concrete is necessary to finish the chamber. For the outer end portion of the dock, where the bedrock dips under the water surface, the walls must be extended to the required height by concrete masonry side walls. These concrete side walls are to be anchored to the bedrock by means of sections of railway rails used as holding-down bolts and set every 10 feet apart.

Referring particularly to the dock chamber construction, the entire chamber will be concrete lined except for the sides of the approach, the seat for the caisson and the apron arch, which will be of cut granite masonry. There are to be sixteen altars on each side of the chamber extending from a point 100 feet from the gate seat to a point within 150 feet of the head of the chamber. The bottom of the chamber will have a timber flooring embedded in concrete to support the keelson, bilge ways and bilge blocks.

The concrete side-wall lining begins at a point at least 1 foot below the surface of the rock bottom, and is carried up along the sides. Molds are to be used to give the proper form and dimensions to the lining, and the concrete is to be rammed into these molds in 6-inch layers. Between the concrete lining and the rock surface there is to be a waterproof coating consisting first of a coat of refined asphaltum reduced in benzine and sprayed on, and then of two coats of asphaltum applied hot. The entire surface of the concrete lining is to be plastered with

a ½-inch coat of one cement and one sand mortar floated and troweled to a hard, smooth surface. All concrete veneers will be additionally strengthened against checking with an interstitial web of expanded metal.

In building the floor the rock bottom of the dock chamber is to be cleaned and recessed for the sill or bilge-way timbers. These sills are to be 12 by 12-inch Douglass fir timbers laid transversely across the dock 10 feet apart and recessed 6 inches into the rock bottom, to which they are to be fastened by 7 feet by 1⅛-inch anchor bolts spaced 7 feet apart. Running longitudinally of the dock between the sills there are to be 6 by 6-inch stringers spaced 8 feet apart and toe-spiked flush with the tops of the sills. The space between the sills and stringers is to be filled flush with concrete. The keelson extending longitudinally along the center line of the dock is to consist of four 12 by 12-inch yellow fir timbers screw-bolted together and anchor-bolted to the rock bottom. The flooring is to consist of 4 by 12-inch yellow fir planking boat spiked to the stringers for the spaces between sills and of 4 by 18-inch Port Orford cedar plank or bilge ways for the spaces over the sills. These bilge ways will be boat spiked to the sills and have iron guides for the bilge blocks, which are to be of laurel. On each side of the wooden flooring and at the feet of the side walls there will be a gutter built of concrete and surfaced like the side walls. Each bilge block is provided with an eye bolt, and directly opposite in the side walls are pulley blocks, these attachments being provided for the rope used in hauling the bilge blocks to one side. Belaying pins and locating pins are set into the concrete of the curb and the tread of the first steps below.

The concrete used for the lining is to be composed of 18 cubic feet broken stone, 4 cubic feet of sand and 1 barrel of Portland cement, and that for the large walls of 20 cubic feet of broken stone, 5 cubic feet of sand and 1 barrel of Portland cement. The specifications require the mixing to be done as follows:

The crushed rock must first be spread on a wooden platform in layers not to exceed 6 inches in thickness; over this will then

be spread first the sand and then the cement, each in its proper proportion; the bed thus formed must be turned with shovels three times while dry and once after being wet, but under no circumstances will a so-called patent or machine mixer be allowed to be used.

The stone masonry is to be of first-class granite ashlar work.

The entrance of the dock is closed in the usual manner by a steel caisson. This caisson is 104 feet $8\frac{7}{16}$ inches long, with a molded breadth of 22 feet and a depth of 33 feet 9 inches from bottom of keel to the under side of the upper deck. Both basic and acid open hearth steel will be used with the following general physical requirements:

Character.	Phos. *per cent.*	Sulph. *per cent.*	Tens. Str. *Max. pounds.*	Elong. *per cent.*	Reduc. *per cent.*
Basic steel.....	0.05	0.04	65,000	25	45
Acid steel.....	0.07	0.05	70,000	22	40

The caisson is reversible and is divided into six watertight compartments. The dock is filled through the caisson by means of thirteen 30-inch gate valves. Sea valves, 6 inches in diameter, are provided for flooding the caisson, and it will be emptied by a 3,000-gallon centrifugal pump operated by an engine and boiler carried on the caisson. The usual air ports, scuppers, capstans and other appurtenances are provided.

The power plant consists of a boiler and engine house built of brick, and respectively 40 by 90 feet and 50 by 60 feet in plan. The boiler room has plastered walls and a concrete floor, and the engine room has walls, floor, ceiling and roof timbers of Oregon pine with natural wood finish. Along one front of the boiler house there is a row of coal bunkers.

Boilers.—There are to be seven water-tube boilers; six of 200 H.P. each, set in three batteries of two each, and one 75 H.P., one feed-water heater, two duplex feed pumps. Each boiler to be not less than 200 nominal horsepower (one horsepower equals 30 pounds of water evaporated from 60 degrees Fahrenheit to steam at 70 pounds pressure) and to have not less than 2,000 square feet of heating surface (1,800 square feet to be in tubes between headers) and 40 square feet of grate surface. They are to be of the inclined straight water-tube type.

Engines.—There will be three 24 by 48-inch horizontal box frame, heavy duty, Corliss engines, to run at 70 revolutions per minute and develop 345 H.P. at $\frac{1}{5}$ cut-off at 100 pounds steam pressure. Each engine will be connected to pump with a special laid endless rope $1\frac{1}{4}$ inches in diameter. Fly-wheels shall be made in sections.

Pumps.—The main pumps, three in number, will be centrifugal double-suction, each having a 38-inch discharge and 26 inch suction pipes. The runners are to be 60 inches closed, and of three-vane type, operating at a speed of 200 revolutions a minute. Each pump must have a capacity of 37,000 gallons per minute while pumping against a head varying from 0 to 30 feet. There will be placed in the pump pit a duplex, double-acting steam piston pump 12 inches in diameter. This pump will have a capacity of 1,200 gallons per minute under a 40-foot head. The drainage pump will be properly connected with the 75-HP. water-tube boiler.

Pump Pit.—The pump pit will be located where shown on plan. It will be 18 feet wide, 56 feet long, and 37 feet deep, all inside measurements. The floor will be supported on an arch of concrete 2 feet 3 inches in thickness, and will have in addition twelve 20-inch I-beams, 64 pounds to the foot, for supporting pumps. The sides of the pit will be of concrete 12 inches in thickness. The ropeways will be of the dimensions shown, with concrete sides 9 inches in thickness and bottom of same 6 inches thick. The sides of pit and ropeways to be plastered with equal parts of sand and cement, trowled and floated to a smooth and even surface. The sides of walls to be attached to original rock, and all spaces to be filled with concrete.

The interior of pump pit and ropeways, when thoroughly and completely dry, to be painted four coats with pure white lead and boiled linseed oil. The gate valves will be placed on each side of the pump pit in the suction tunnel where shown on plan. —" Engineering News."

A 150-HORSEPOWER THREE-STAGE COMPRESSOR.

A 150-horsepower compressor, designed to compress carbonic-acid gas to a pressure of 1,200 pounds per square inch,

has just been built by the McKiernan Drill Company, of New York City, from the plans of Mr. Charles H. Sergeant, its engineer. The machine is a horizontal steam-driven three-stage compressor and posesses a number of interesting features. Among these may be mentioned, in brief, a means of carrying leakage from the intermediate and high-pressure cylinders to the inlet of the low-pressure cylinder, a ready means for examining the second intercooler, if desired, a joint arranged to hold its packing fast, and accessible valves and cylinder heads. The cylinders, heads, and first and second intercoolers are, of course, water-jacketed, and caps or plates are provided for removing any sediment in the parts. As a straight-line machine for this pressure, it is certainly a very large one.

The steam cylinder is 22 inches in diameter, with 24-inch stroke, and it is designed to develop full power with steam at 60 pounds pressure and a speed of 70 revolutions per minute. The steam chest is above the cylinder, a double-slide valve is used, driven from two independent eccentrics, and speed regulation is by fly-ball governor. The three air cylinders are arranged in line, the low-pressure, which is double acting, next to the steam cylinder, and the intermediate and high-pressure cylinders, both of which are of the single-acting type, beyond, with their abutting ends connected together. The air cylinders are 16, 11 and $4\frac{1}{2}$ inches, respectively, in diameter. The compression valves are of the poppet type and are arranged to fall naturally to their seats.

The first intercooler is of the type employing a number of small tubes through which the air is passed, the working pressure of 75 pounds above the atmosphere admitting of this usual construction. In the second intercooler a comparatively long, single 5-inch pipe is used, the slow passage of air through it providing for the cooling. The pipe is enclosed in the water-cooling chamber.

The high-pressure connections required in these cases are fitted with asbestos packing rings which are pressed between the flange and the casting on which it is seated. The rings lie in an annular cut in the casting and are held on the outside by

the casting and on the inside by the projection of the flange, this being continued at reduced diameter to slip within the opening to be connected. As was mentioned, the intermediate and high-pressure cylinders are connected together, and at this point, the leakage pipe is taken to the low-pressure cylinder, catching the air which may leak by the intermediate and high-pressure piston rings.

A short test of the machine was made with steam supplied from a 25-horsepower boiler equipped with an automatic powdered-coal burning apparatus, and generating steam at 75 pounds pressure. The discharge of the compressor was led into a 5-inch, double-extra-strong pipe, 10 feet 6 inches long, in which the air was brought to a pressure of 1,500 pounds in about 4 to 5 minutes, with the machine working at an average rate of 40 revolutions per minute.—"Engineering Record."

THE NAVAL ARCHITECTS AND MARINE ENGINEERS.

The eighth annual meeting of the Society of Naval Architects and Marine Engineers, which was held in New York City, November 15 and 16, was one of unusual interest. One evidence of the estimation in which the proceedings of this society are held abroad was in the presence at the meetings of Lieutenant Commander De Faranaud, of the French Navy; Lieutenant Commander Von Rebeur Paschwitz, of the German Navy; Captain Stohensnovitch, Imperial Russian Navy; Captain Beher, Imperial Russian Navy, and Naval Constructors Tehernigvosky and Makedonsky, Imperial Russian Navy.

President Clement A. Griscom opened the meeting with an address, dwelling on the money now being expended upon naval construction in this country, the improvements in design and construction and the increased interest in naval matters. F. Merriam Wheeler read a paper describing a capacity test of an air pump used in connection with surface condensers. A paper by W. D. Forbes was on "The Interchangeability of Units for Marine Work." Duplicate or interchangeable parts being concededly so valuable, it was argued that the principle could be

advantageously extended to embrace larger units or complete apparatus. Naval Constructor Taylor's paper described the United States experimental model tank. Professor A. H. Sabin read a paper on "The Composition and Classification of Paints and Varnishes;" Naval Constructor Woodward described "Tests of the Electric Plants of the Battleships *Kearsarge* and *Kentucky;*" and Spencer Miller told of "The Coaling of the *Massachusetts* at Sea."

The morning session of the second day of the meeting was presided over by Charles H. Cramp, and two important papers were attentively listened to. Assistant Naval Constructor H. S. Gilmor described "Recent Improvements in Foreign Shipbuilding Plants," and George N. Dickie discussed the question "Can the American Shipbuilder Compete with the British and German Shipbuilders in the Production of the Largest Class of Ocean Passenger and Freight Steamships?" Among other things Mr. Dickie said:

"I have lately had to discuss with the chief surveyor at Lloyds' certain departures, that we considered slight and in the line of improvement, from the rules as applied to two large freight twin-screw steamers we are now building. I have found that the skill of the experts at Lloyds', and the careful study of every proposition made to them, not only protects all parties, but insures a steady forward movement. I believe, however, that an American register of shipping that would be accepted by the shipowner and by the underwriters the world over, would be a great help to the American shipbuilder, as it would no doubt lend itself more readily to the tendencies of American design. This would bring about a closer adaptation of the material used to its place and function in the complete design. In this the steel manufacturer will likely take a share.

"We are on this point now abreast of the British builder, and, in methods of handling, ahead of most of them. In skill, I think the American naval architect is quite abreast of his British cousin, and, if he has plenty of practice, there need be no fear of keeping him in that position."

On the relative cost of labor in American and British shipyards, Mr. Dickie said:

"From combining what information I have been able to gather I find that on an average the steel work of construction costs in the British yards from £3 17s. to £4 per ton of material worked. This, I think, we can about equal in labor cost here. We employ the same class of men as they do. Our total labor cost is about 25 per cent. greater than theirs. As to what we may be able to do in regard to equalizing the cost of labor, I do not think it will be brought about by any attempt to equalize the rate of wages paid throughout the shipbuilding world."

"I find that the marine engine and its boilers cost less in proportion to the hull in England than they do with us. On the whole, I think the British builder has an advantage over the American builder in the cost of material. This applies even to the wood used. As near as I can judge at the present time, there is 10 per cent in cost of material in favor of the British shipbuilders."

The following papers presented at the afternoon session we can give only the titles of: "Recent Designs of Battleships and Cruisers of the United States Navy," by Chief Constructor Philip Hichborn; "A Comparison of the Contract Prices of Our Naval Vessels," by Harrison S. Taft; "Launch of a Cruiser," by James Dickie; "The Safety of Torpedo Boats at Sea and in Action Under Various Conditions," by Naval Constructor Lloyd Bankson, and "Classification of Rules," by Theodore Lucas.

The meeting closed with a banquet at Delmonico's, at which about 150 members were present.—"American Machinist."

A SPEED TEST OF A WOOD AND IRON PULLEY.

A pulley with iron center or spider and wooden rim, 46½ inches diameter and 16 inches face, was recently ordered by the Crocker-Wheeler Electric Co., of Newark, N. J., to be fitted to a 4-inch shaft, and it was required that it should be tested to a rim speed of 9,000 feet per minute, that being the service speed of the pulley. The order was placed with the Dodge Mfg. Co., of

Mishawaka, Ind., and that company concluded, not only to make the required test, but to test a duplicate pulley by running it at very much higher speeds, and, if possible, to the point of bursting. The rims of the two pulleys were made from the same lumber, and the iron spiders were molded on the same day and cast from the same ladle. The service pulley was tested to a rim speed of 10,000 feet and shipped to the purchasers.

The plant for driving the test pulley was then installed in an unused room, connection being made with a line shaft, and barricades erected to prevent any accidents from the bursting of the pulley. Connection was made with the main engine by rope transmission from the main jack shaft to a line shaft on which was a 60-inch pulley bolted to a 14-inch all-wood pulley on the countershaft. This countershaft carried a 48-inch wooden pulley bolted to a 16-inch pulley of the same construction and mounted on the same mandrel as the test pulley. The test was made during the dinner hour, when the engine could be run at varying speeds. It was started slowly and gradually increased to its maximum speed of 75 revolutions per minute, giving the line shaft 200 and the countershaft 857 revolutions.

Readings were taken from the test pulley until the engine reached its maximum speed, which would theoretically have given the pulley a speed of 2,571 revolutions.

The readings, however, indicated a loss of about five per cent. in the transmission, the actual speed being a little over 2,400 revolutions per minute. This gave a rim speed of 29,228 feet, or over 5½ miles per minute. There was no perceptible weakening of the pulley after this test, but it is proposed to make arrangements for higher speeds in order to ascertain the bursting speed. Such a speed as above noted, however, is far beyond what is practicable with ordinary cast-iron pulleys of the same size.

It may be noted that the tensile stress per square inch of cross section of the rim at the above-named speed would be about 1,700 pounds, assuming the wood to weigh 37.5 pounds per foot, and this is so far inside the breaking strength of sound wood that bursting could hardly have been expected at this

speed. A cast-iron pulley of this diameter, run at such a speed, would have a rim tension of about 20,500 pounds per square inch, and would be apt to give way before reaching such a speed on account of the bending of the rim, due to the action of the pulley arms when the rim stretches under the tension due to high speed. The reason why wooden-rim pulleys can run at so much higher speed than cast iron is their small weight and their elasticity, which enables them to yield without injury to the bending stresses which occur at high speed.—"Engineering."

NOMENCLATURE OF ELECTRICAL APPARATUS.

According to the direction of the energy transmitted, electric machines have been divided into generators and motors. By the character of the electric power, they have been distinguished as direct-current and as alternating-current apparatus.

With the advance of electrical engineering, however, these subdivisions have become unsatisfactory and insufficient.

The division into generators and motors is not based on any characteristic feature of the apparatus, and thus is not rational. Practically any electric generator can be used as a motor, and inversely, and frequently one and the same machine is used for either purpose. Where a difference is made in the construction it is either only quantitative, as, for instance, in synchronous motors a higher armature reaction is used than in synchronous generators; or it is in minor features, as continuous-current motors usually have only one field winding, either shunt or series, while in generators frequently a compound field is employed. Furthermore, apparatus have been introduced which are neither motors nor generators, such as the synchronous machine producing wattless lagging or leading currents, etc., and the different types of converters.

The subdivision into continuous-current and alternating-current apparatus is unsatisfactory, since it includes in the same class apparatus of entirely different character, such as the induction motor and the alternating-current generator, or the con-

stant-potential commutating machine, and the rectifying arc-light machine.

The following classification, based on the characteristics features of the apparatus, has been adopted by the A. I. E. E. standardizing committee in its report. It refers only to the apparatus transforming between electric and electric, and between electric and mechanical power:

1st. *Commutating Machines*, consisting of a uni-directional magnetic field and a closed-coil armature, connected with a multisegmental commutator.

2d. *Synchronous Machines*, consisting of a uni-directional magnetic field and an armature revolving relatively to the magnetic field at a velocity synchronous with the frequency of the alternating-current circuit connected thereto.

3d. *Rectifying Apparatus*, that is, apparatus reversing the direction of an alternating current synchronously with the frequency.

4th. *Induction Machines*, consisting of an alternating magnetic circuit or circuits interlinked with two electric circuits or sets of circuits, moving with regard to each other.

5th. *Stationary Induction Apparatus*, consisting of a magnetic circuit interlinked with one or more electric circuits.

6th. *Electrostatic* and *Electrolytic Apparatus*, as condensers and polarization cells.

Apparatus changing from one to a different form of electric energy have been defined as:

A—*Transformers*, when using magnetism, and as

B—*Converters*, when using mechanical momentum as intermediary form of energy.

The transformers, as a rule, are stationary; the converters, rotary apparatus. Not included under converters are motor generators transforming from electrical over mechanical to electric power by two separate machines, and dynamotors, in which these two machines are combined in the same structure.

1. *Commutating Machines* as *Generators* are usually built to produce constant potential for railway, incandescent lighting and general distribution. Only rarely are they designed for ap-

proximately constant power for electro-metallurgical work, or approximately constant current for series incandescent or arc lighting. As *Motors* commutating machines give approximately constant speed—shunt motors—or large starting torque-series motors.

When inserted in series in a circuit and controlled so as to give an e. m. f. varying with the conditions of load on the system, these machines are *Boosters*, and are generators when raising the voltage, and motors when lowering it.

Commutating machines may be used as *Continuous-Current Converters* by transforming power from one side to the other side of a three-wire system.

2 While in commutating machines the magnetic field is almost always stationary and the armature rotating, *Synchronous Machines* are built with stationary field and revolving armature, or with stationary armature and revolving field, or as inductor machines with stationary armature and stationary field winding, but revolving magnetic circuit.

According to the number and character of the alternating circuits connected to them, they are single-phase or polyphase machines. As *Generators* they comprise practically all single-phase and polyphase alternating-current generators; as *Motors*, a very important class of apparatus, the *Synchronous Motors*, which are usually preferred for large powers, especially where frequent starting and considerable starting torque are not needed. Synchronous machines may be used as *Compensators* to produce wattless currents, leading by over excitation, lagging by under-excitation, or may be used as *Phase Converters* by operating a polyphase synchronous motor by one pair of terminals from a single-phase circuit. The most important class of converters, however, are the synchronous commutating machines.

Synchronous Commutating Machines contain a uni-directional magnetic field and a closed-circuit armature connected simultaneously to a segmental continuous-current commutator and by collector rings to an alternating circuit, mostly a polyphase system. These machines thus can either receive alternating and yield continuous-current power as synchronous converters, or

simply "*Converters*," or receive continuous and yield alternating-current power as *Inverted Converters*, or driven by mechanical power yield alternating and continuous current as *Double Current Generators*. Or, they can combine motor and generator action with their converter action. Thus a common combination is a synchronous converter supplying a certain amount of mechanical power as synchronous motor.

3. *Rectifying Machines* are apparatus which by a synchronously-revolving rectifying commutator send the successive half-waves of an alternating single-phase or polyphase circuit in the same direction into the receiving circuit. The most important class of such apparatus are the *Open-Coil Arc-Light Machines*, which generate the rectified e. m. f. at approximately constant current, in a star-connected three-phase armature in the Thomson-Houston, as quarter-phase e. m. f. in the Brush arc-light machine.

4. *Induction Machines* are generally used as *Motors*, polyphase or single-phase. In this case they run at practically constant speed, slowing down slightly with increasing load. As *Generators* the frequency of the e. m. f. supplied by them differs from and is lower than the frequency of rotation, but their operation depends upon the phase relation of the external circuit. As *Phase Converters* induction machines can be used in the same manner as synchronous machines. Their most important use besides as motors is, however, as *Frequency Converters*, by changing from an impressed primary polyphase system to a secondary polyphase system of different frequency. In this case, when lowering the frequency, mechanical energy is also produced; when raising the frequency, mechanical energy is consumed.

5. The most important *Stationary Induction Apparatus* is the *Transformer*, consisting of two electric circuits interlinked with the same magnetic circuit. When using the same, or a part of the same electric circuit for primary and secondary, the transformer is called a *Compensator* or *Auto-transformer*. When inserted in series into an alternating circuit and arranged to vary the e. m. f., the transformer is called *Fotential Regulator* or booster. The variation of secondary e. m. f. may be secured by

varying the relative number of primary and secondary turns, or by varying the mutual induction between primary and secondary circuit, either electrically or magnetically. The stationary induction apparatus with one electric circuit are used for producing wattless lagging currents as *Reactive* or *Choking Coils.*

6. *Condensers* and *Polarization Cells* produce wattless leading currents, the latter, however, at a very low efficiency, while the efficiency of the condenser is extremely high, frequently above 99 per cent; that is, the loss of energy is less than 1 per cent. of the apparent volt-ampere input.

To this classification may be added the *Uni-polar*, or more correctly, *Non-polar* machine, in which a conductor cuts a magnetic field at a uniform rate. Thus far these machines do not appear of any practical value.

Regarding apparatus transforming between electric energy and forms of energy differing from electric or mechanical energy, the transformation between electrical and chemical energy is represented by the primary and secondary battery and the electrolytic cell, the transformation between electrical and heat energy by the thermophile, and the electric heater or electric furnace and the transformation between electrical and light energy by the incandescent and arc lamps.—CHARLES P. STEINMETZ, in the "Electrical World and Engineer."

ACCIDENT ON THE *GARIBALDI.*

A rupture of tubes occurred in two boilers on board of the Italian armored cruiser *Garibaldi* while on her trial trip July the 12th, causing the death of one coal passer and a severe injury of two firemen. The *Garibaldi* has two engines of 13,500 I.H.P., aggregate, that receive their steam from twenty-four Niclausse boilers each of 12 elements.

The start was made with steam in the twelve forward boilers, while the twelve aft boilers were gotten ready for steam. It was the intention to make the trial progressive, with steam pressure increasing until the maximum power should be reached. One and a half hours after leaving her anchorage, and while the en-

gines were running about 60 revolutions per minute, the aft boilers were connected to the forward boilers that were nearest to the engine, and with these the trial was begun. One quarter hour later an explosion took place in boiler No. 16, by which two firemen were severely burned. Luckily the fire doors were closed so that the steam could escape through the ashpit, by which circumstance further calamities were prevented. When access could be obtained, the aft boilers were shut off, commencing at aft, and the fires extinguished. Twenty minutes after the above-mentioned explosion there followed a second explosion, in boiler No. 13, causing the death of one coal passer.

At the subsequent examination there was found one tube ruptured in each boiler. It also could be seen from the fire doors that the tubes were bent from four to five inches downwards, wherefore it became necessary to take seven rows of tubes out of each of the twelve elements, since they were all bent or injured. The tubes had to be hammered out of their conical sides, as is usual in other cases in service, while in new boilers they fall out themselves. It was found that insufficient water height caused the explosion. After replacing the two injured boilers the *Garibaldi* will again take up her trial. "Mitt. Geb. Seew.," Oct. 26, 1900, 934.

FRENCH NAVAL CONSTRUCTION FOR 1901.

The French ship building program for the year 1901 has just been issued. The names of no less than 111 vessels are included in this statement, but thirteen of these are to be completed during the present year, including the *Iéna*, a 12,000-ton battleship, and the submarines *Farfadet* and *Algérien*. During the year the battleships *Henri IV*, 9,000 tons, and *Suffren*, 12,000 tons, the armored cruisers *Jeanne d'Arc* and *Montcalm*, and the first-class cruiser *Jurien de la Gravière*, will be added to the French navy, as well as the three destroyers *Pertuisane*, *Escopette* and *Flamberge*. Five submarines—the *Sirène*, *Triton*, *Korrigan*, *Gnome* and *Lutin*—will also be completed, besides seven seagoing torpedo boats, and twelve smaller boats. The battleship *Henri IV* is building at Cherbourg, and is to be finished by April

1; the *Suffren*, at Brest, is to be ready by May 1. Two new battleships, known at present as the *A8* and *A10*, laid down under the act for the development of the fleet, are to be of a new type. They will have a displacement of 14,865 tons, 14,475 H. P., 18 knots speed and 8,390 miles radius of action when steaming at 10 knots. Their armament will include four heavy guns, eighteen guns of medium caliber and twenty-eight smaller pieces, with five torpedo tubes. The estimated cost of each is 35,500,000 francs. These ships, it will be noticed, correspond very nearly to the British *Formidable* type.

Of armored cruisers France has in all fourteen under construction, divided into five different types. Of these the *Jeanne d'Arc*, to be completed towards the end of 1901, is the only vessel of her class. She has a displacement of 11,270 tons, and will cost nearly 17,000,000 francs. This vessel, it will be observed, is a little smaller than the English *Cressy* class, but is to have two knots extra speed, 23 knots, as compared with the 21 knots of the English ships. Three other armored cruisers—the *Dupetit Thouars*, *Gueydon* and *Montcalm*—have a displacement of 9,517 tons. They are being constructed at Toulon, Lorient and La Seyne respectively. The speed of this class is to be 21 knots. After these come five cruisers of 10,014 tons—the *Condé*, *Gloire*, *La Marseillaise*, *Sully* and *Amiral Aube*. The two first are building at Lorient, the third at Brest and the other two at Seyne and Saint Nazaire, respectively. There are also three armored cruisers of 7,700 tons, specially designed for service in foreign waters. The twelve cruisers above mentioned were laid down before the close of the year 1899. In 1900 two vessels of a new type were commenced, the *Jules Ferry* and the *Léon Gambetta*, the former at Cherbourg, and the latter at Brest. They are to be of 12,550 tons displacement, of 27,500 H.P. and 22 knots speed. Lastly, an armored cruiser, known as *C13*, similar to the *Jules Ferry*, will be commenced at Toulon during the coming year. With regard to protected cruisers, since it has been decided to discontinue building this type of vessel, there is but one under construction, the *Jurien de la Gravière*, now building at Lorient. She is to be completed by October 1.

A distinctive feature of the new French program is the entire absence of torpedo boats, either of the seagoing or harbor type, these having been given up in favor of the destroyers, which it is said "can keep the sea better and act more advantageously as scouts." Thus there remain under construction only the seven seagoing torpedo boats previously mentioned, and these will be completed in 1901. On the other hand, the building of destroyers will be pushed on energetically, fourteen being in hand, while ten others are to be laid down. These will be divided between the government and private yards. Much interest naturally centers in the construction of submarines. Altogether seventeen such vessels are provided for. Two of these, as has already been mentioned, are to be completed in 1900. The *Triton* and *Sirène*, building at Cherbourg, which were delayed in construction, pending the completion of the trials of the *Narval*, have lately been pushed forward. Two new submarines of the submersible type—the *Silure* and *Espadon*—have been commenced at Cherbourg during the current year. During 1901, eight others will be put in hand, six at Toulon and two at Cherbourg. The complete flotilla of submarines, as at present arranged, will therefore consist of twenty-two vessels, of which seven will be completed in 1900, the same number in 1901, five in 1902, and three in 1903.—"Marine Review."

TWENTY-FOOT BOILER TUBES FOR LOCOMOTIVES.

For several years there has been a tendency toward lengthening locomotive boiler tubes, and it seems likely to receive considerable impetus through the influence of wider fire boxes for bituminous coal. The tendency is not only to use longer tubes, but to increase the ratio of length to diameter, and one motive-power officer writes that he has changed the tubes in a mogul engine from 2 inches to $1\frac{3}{4}$ inches without changing the length, which was 12 feet 6 inches, and has "done wonders with them." He is also using $2\frac{1}{4}$-inch tubes, 16 feet long, and has no fear of them.

Sixteen feet is common enough now to cause no comment

when this dimension appears in a new engine. Mr. S. M. Vauclain advocates 2-inch tubes 20 feet long, and at the present time a length of 19 feet has been adopted in a new design for fast passenger service, the diameter in this case being 2¼ inches. We think that there are enough straws to indicate which way this wind is blowing. The wide fire box has raised the question of the length of tubes, because of its effect upon wheel arrangements. The desire to use long tubes is increased by the necessity of getting large wheels in front of the mud ring.

The relation between the length and diameter is most important, and it is to this that special attention should be directed. The length should not be increased without consideration of this ratio, and there are good reasons for believing that an increase in the prevailing ratio is desirable.

In European (Continental) practice the ratio has been 60 for express engines, it has been about 75 in English practice, and from 70 to 80 in American, although Mr. G. R. Henderson has suggested the limits of 70 and 90.

The Pennsylvania Class E1 locomotive was designed with a view of using a ratio of 90, although 86 was finally employed. This shows the tendency towards increasing the ratio and a new design, for which drawings are now completed, will use 19 feet tubes with a ratio of 100. In Russia 2-inch tubes are in use with a ratio of 108, the length being 18 feet 1 inch, and if Mr. Vauclain should put his idea into practice we shall have a ratio of 120 with 2-inch tubes 20 feet long.

The famous experiments of M. Henri, chief engineer of the Paris, Lyons and Mediterranean ("American Engineer," August, 1890, page 337), and the opinion based upon them has unquestionably influenced locomotive practice in confining tube lengths in general practice between 12½ and 14 feet. These tests showed a gain of 7 per cent. in evaporation by an increase from 13 to 16 feet in length without changing the diameter and tubes; 23 feet long gave 30 per cent. more evaporation than those of 10 feet. The 23-foot tubes, however, increased the draft resistance. These tests, however, used drafts of 1 to 2.95 inches, and drafts of 14 or 16 inches, such as occur in American practice, might change

the conclusions entirely. M. Henri used 78 pounds of coal per square foot per hour as a maximum rate of combustion. When 250 or 300 pounds are burned per hour the velocity of the gases in the tubes is vastly greater, and therefore these experiments now seem to point toward the desirability of much longer tubes in this country, at least that is the view taken by several well-known men.

Increased friction and reduced draft effect will undoubtedly result from increased lengths of tubes, but we believe that there is much more to be gained in the greater heat absorption than will be lost in these ways. Two different tubes with the same ratio of length to diameter will give the same efficiency for the same velocity of the gases, and the long tube may be made larger if necessary. Even if it causes a slight sacrifice in amount of heating surface it is possible that the greater length will be more advantageous than slightly more heating surface in shorter tubes. To determine this positively a very difficult test must be made.

The influence of velocity of the gases has been referred to before in these columns in connection with Wöhler's mathematical analysis of Henri's experiments. The velocity of the gas current affects the action between the gas and the heating surface as does the temperature. The period of contact of the molecules varies inversely as the velocity of the current. Wöhler has worked out a table (Bulletin of the International Railway Congress, June, 1899, page 820) to show his idea of the proper ratio of length to diameter required to obtain different degrees of efficiency when the velocity of the gases varies from 5 to 20 feet per second. These figures call for a ratio of 100 when ordinary efficiency is expected and of 130 as a maximum when the velocity is 20 feet per second. It is our opinion that in recent practice the velocities greatly exceed 20 feet per second. The higher the velocity the longer the tubes should be, and for locomotives with high rates of combustion there is good reason to believe that the tubes cannot be too long within the limits imposed by restrictions of weight and space.

M. Henri proved that with light drafts 23-foot tubes gave an advantage of 12 per cent. over 13-foot tubes in water evapora-

tion per pound of coal. We may yet come to the 23-foot tubes, but before this point is reached the Serve tube should come up for consideration. For a given diameter Serve tubes have a heat-absorbing surface 75 per cent. greater than that of ordinary tubes, but they are expensive.

There seems to be but one anxiety in the use of long tubes, that concerning the expansion and contraction and its effect upon leakage at the tube sheets. This fear may prove to be without foundation, but if not, there is a simple remedy in cambering the tubes by giving them a slight bend before being placed in the boiler. Cambering has been practiced for five years on the Caledonian Railway of Scotland, and we are informed by Mr. J. F. McIntosh, locomotive superintendent of that road, that it was inaugurated for the purpose of relieving the tube sheets from these effects, and also to increase the resistance of tubes to bending by their own weight and thereby lessen the injurious effects of vibration in producing leakage at the tube ends. In the opinion of Mr. McIntosh, when the tubes are cambered they are more flexible longitudinally, and, therefore, yield more freely to the expansion and contraction, and reduce the stresses at the tube sheets. The tubes are generally cambered by the manufacturers, but this has been satisfactorily done in the shops of the Caledonian in a screw press.—"American Engineer and Railroad Journal."

THE PRODUCTION OF MANGANESE.

The demand for manganese in recent years in connection with the manufacture of Bessemer and open-hearth steel has had the natural effect of attracting capital to ore deposits in many parts of the world. As it happens, this element is tolerably widespread in its occurence, and there is no reason to apprehend any failure in the supplies for many generations to come. Formerly the world's supplies were mostly derived—and in relatively small quantities, too, because of the limited demand then existing—from Spain and from certain districts in Thuringia, North Hesse and Moravia. But now, Spain is a poor second to Russia as a source of supply, and Germany ranks among the

insignificant producers. It is estimated that the aggregate output for last year was about 800,000 tons of ore, of which 369,000 tons came from Russia, 140,000 tons from Spain and Portugal, 77,000 tons from India, 62,000 tons from Brazil, 59,000 tons from Germany, and the balance from minor sources. In the immediate future the main dependence will continue to be upon Russia, which is capable of progress even more marked than that of the last few years. Poti is the chief shipping port, but a few thousand tons are sent away annually by way of Batoum. In 1898 the exports from these two places, including about 30,000 tons shipped to other parts of Russia, were 282,316 tons, which compares with 215,263 tons for 1897. Some portion of the increase of 153,737 tons in two years is referable to better and cheaper means of transport, more especially between the stations of Tchiaturi and Sharopan. But for the inadequate harbor accommodation at Poti, more than 400,000 tons might have been sent abroad last year. As it is, there is only a very limited space for shipping at that port, and the conveyance of the manganese by lighters to steamers in the roadstead is a rather costly and very slow process.

In his last report, Mr. Consul Stevens suggested that the existing unsatisfactory state of affairs would be considerably improved if a reduction were made by the State railway authorities in favor of the transport of ore to Batoum, where greater facilities in the shape of harbor accommodation exist, and in consequence of which vessels would charge a lower rate of freight for the conveyance of the ore by sea. The existing disability causes production to be not over profitable for the producers, who do not appear to be able to reap the benefit of an advance in price in foreign countries, notwithstanding the fact that it costs them next to nothing to procure the ore at the mines, and that a reduction has taken place on the cost of transport over the narrow-gage railway as far as the trunk line.

A country which is cultivating a trade in manganese is Brazil, a newcomer of much promise. In a paper read a few months ago before the Iron and Steel Institute, Mr. H. K. Scott went into considerable detail on the subject, with particular reference

to the mines which have been opened up since 1894. The descriptions found in greatest abundance are manganite and pyrolusite, minerals of excellent quality. Only deposits in the neighborhood of the railway are worked at present, the principal workings being situated at Queluz and Miguel Burnier, at 462 and 496 kilogrammes, respectively, from Rio de Janeiro. The mineral is worked in open cuttings and the cost of extraction is said to be about 10s. per ton. Very fair deposits also occur at Gandarella, Minas Geraes. Mr. Scott tells us that the ore is a very pure black oxide, containing from 50 to 55 per cent. of metallic manganese, and only traces of phosphorus. The ore is first excavated by open-cast workings, and when these are no longer practicable, they are superseded by levels in the hillside, small levels being first driven to locate the direction of the curving and erratic beds of ore. All the levels and workings are dry, owing to their altitude, and the ore is simply thrown into the shoots without the necessity for picking, the cargoes showing a general uniformity in their analysis. The amount of ore in sight, and of easy extraction without motive power, is estimated by Mr. Scott at 2,000,000 tons on the present Usina Wigg properties. The railway rate to Rio has varied from about 3s. 8d. to 9s. 4d. per ton, but is now about 5s.; whilst the freights to England vary from 10s. to 20s.

There are other districts in which manganese ores are, or have been, mined since 1895; but the majority of them are described as valueless from a commercial point of view, owing to excessive impurities, some being too phosphoric and others too silicious. A few consist of the lower oxide of manganese, contaminated and blackened with a carbonaceous material; whilst occasionally the surface ore is merely a cap, and not a bedded deposit. Still, there is no question of Brazil's ability to meet, or to help materially to meet, the growing demand for manganese. On the other side of South America we have Chile, which exported 23,000 tons last year, bringing the total for 11 years to 367,000 tons, three-fourths of it coming to England. Nearer home we have undeveloped resources in Germany, in spite of the drain of the last few years. Belgium, too, which produced

16,440 tons last year, appears to be good for an increase. The same remark applies to Sweden, which last year raised 2,358 tons of ore, and 363,000 tons of pyrolusite. The mines are not very ready of access, which fact naturally interferes with development.

Of last year's output of manganese ore throughout the world, about 600,000 tons were used up in Europe, and 200,000 tons in the United States. America, it may be observed, is going more to Cuba for its requirements in this respect than it did during the Spanish administration of the island. Cuba, in fact, should, when properly exploited, meet the whole of America's requirements. In the province of Santiago de Cuba there are 88 manganese mines covering 4,015 hectares. The deposits lie in the San Maestro range on the south coast, extending over a distance of 100 miles between Santiago and Manzanillo.

As the demand in the United States for manganese was far in excess of the native supply, and the nearest known mines were in the neighborhood of the Black Sea in Europe, and in the northern part of South America, attention was, some years back, drawn to the Cuban deposits, and one American company was formed, known as the Panupo Iron Company, to work ground 16 miles north of Santiago, with a railroad extending to that port. Other companies also began work, and the shipments from 1890 to 1893 inclusive amounted to 62,601 tons. In 1894 there was none, and in 1895–6 the total shipments were 750 tons. This decrease in business was due in some measure to low prices, and to other causes than the insurrection and war, but this last was the prime cause of the decrease. Mining has now been resumed with every prospect of continued increase and prosperity. The chief properties are the Macio, on the sea coast, 35 miles from the port of this name; the Portillo, situated further westward, which produces ore that gives only 36 per cent of manganese; the Boniato, Dos Bocas Margarita, Avispero (which yield ore of remarkably fine quality running to 56 per cent of metallic manganese), San Juan, and numerous other groups which could be made very productive and very profitable with the expenditure of only a little capital. Since the Spanish

expulsion, however, the Americans have been acquiring mines, and in all likelihood they have bought up the best on the market.—"Engineering."

PATENTS: WHAT THEY ARE NOT.*

By Paul Synnestvedt.

It is not the purpose of this paper to present a lenghty treatise on patents, from a strictly legal standpoint, but rather to try and give to the members of the Western Railway Club a little instruction from the standpoint of a railway man, which the writer, from almost constant association with railway men and railway work, still feels himself to be, although for some years past his more immediate occupation has been the practice of patent law.

But few of our busy railroad men have had opportunity or occasion to apply their minds to studying the subject of patents, and it necessarily follows that but a very small proportion have any proper conception as to the nature of a patent right; or, to put it differently, what a patent is; or, more accurately still, perhaps, what it is *not*, what relation one patent bears to another, and whence arise so many apparently anomalous situations as are encountered under our patent law.

Most men who are blessed with at least ordinarily astute minds naturally suppose that when an inventor takes out a patent he gets thereby a right to proceed unmolested with the manufacture, sale and use of his invention. That is not the case, however, as we shall see from a little investigation.

The origin of the erroneous idea above stated may, perhaps, be traced to the language employed in the patent grant itself, and in the constitutional clause which is really the basis of the patent system.

It was provided in the Constitution that Congress should have the power "to promote the progress of science and useful arts, by securing for limited times to authors and inventors the exclusive right to their respective writings and discoveries." Under this provision of the Constitution, laws were passed providing

*A paper recently read before the Western Railway Club.

for the grant of patents. The foundation upon which such laws are built involves the idea of a contract between the inventor and the public. In exchange for a full and complete disclosure of the invention, to be preserved in the public records, and accessible at all times to the public, the Government agreed to undertake to secure to the inventor the *exclusive* right to his invention for a term of years. From the beginning, the language employed in the patent itself followed the wording of the above constitutional clause. An example, is taken from what is said to be the first patent ever issued by the United States. It was granted January 31, 1791, to Francis Bailey, of Philadelphia, and related (to quote the patent itself) to certain "Methods, not before known or used, for forming punches, by which to impress on the Matrices of Printing Types, whether such Types be for Letters or Devices, as well as to impress on any Metal or other substance capable of receiving and retaining impressions, various Marks which are difficult to be counterfeited."

The grant recites that "the said Invention appears to be useful and important," and that "in pursuance of the Act entitled, 'An Act to promote the progress of useful Arts,'" there is granted "to the said Francis Bailey, his Heirs, Administrators and assigns, for the Term of fourteen years, the sole and exclusive Right and Liberty of using and vending to others the said Improvement, according to the true Intent and Meaning of the Act aforesaid."

The essential features of the grant have not really been changed since the foundation of the patent system, over one hundred years ago.

Examination of the terms of a modern grant will show that it does not differ materially in substance from the early one, except, perhaps, in the recital of the several steps taken by the inventor to procure the patent. It will be noticed that this grant also purports to convey "the exclusive right to make, use and vend said invention throughout the United States and the Territories thereof."

As a matter of fact, neither of the above grants give the inventor the right to make, sell, or use his own invention, but only the right *to prohibit or prevent others* from making, using,

or selling his invention for a definite number of years, for the infringement of which right he may, under the law, recover damages or profits from the infringer.

That by the grant of a patent the Government does *not* give the inventor the right to make, sell or use his own invention, is evident from the fact that prior to such grant *he already has such right*, provided there are no patents to earlier inventors which he infringes; and in case such other patents or conflicting rights exist, the mere issue of a patent to him will not relieve from the charge of infringement any attempt to make, use or sell his patented device, whether such attempt be made by him or anyone else.

It is the word "exclusive" that really gives character to the grant—the right to *exclude* or *prohibit* others from doing something. Whether a patentee has a right to operate under his own patent or not is entirely dependent upon the existence or non-existence of prior claims held by others, which would be infringed by such operation; and this is a question entirely different from the question as to whether this particular patentee's rights are valid, or infringed by later inventors.

What has been said concerning the nature of the patent grant will, perhaps, help to explain what so many have difficulty in understanding, *i. e.*, how it can be possible for more than one to hold what appears to be a valid patent upon substantially the same thing. As a matter of fact, that is *not* possible; it is only an appearance. The difficulty generally arises in a case where one man holds what is known in patent law as a *broad* or *generic* patent upon a certain invention which has been improved upon by others in various ways, the others securing patents upon their several *improvements*. The man who holds a broad or generic patent has a right to prohibit its use by everyone else, so long as his grant continues alive; but he has *not* the right to prevent or prohibit others from exercising their inventive faculties in the development of *improvements* upon his invention, nor has he the right to prevent or interfere with others securing patents upon such improvements. That would not "promote" the progress of science and the useful arts, but manifestly *retard* it. As has

been well stated by the United States Supreme Court, the disclosure of a broad, generic or pioneer invention not only does not stop or check development along the same line, but rather serves to stimulate it.

To illustrate the distinction between what is known as a generic and what is known as a specific patent, and the rights of the parties holding the same, let us take, as an example, the case of a car coupler. Suppose A invents an improved vertical plane coupler or drawbar, comprising essentially three parts—a *head*, a *knuckle* and a *locking pin*. Suppose he is the first who has ever employed such three parts in combination in a coupler. He is entitled to, and can procure, a patent upon the combination between a head, a knuckle and a locking pin, his claim being entitled to the broadest interpretation by the courts.

Suppose B now takes a coupler made in accordance with A's invention, and, in using the same or studying upon it, works out a different form or arrangement of the locking pin and knuckle. B is entitled to procure, on the filing of proper papers, a patent on his invention, claiming his specific or particular improvement on A's generic invention. The existence of A's patent has not had, and obviously should not have, any effect at all in preventing B from securing a perfectly valid patent on the specific improvement which he has invented; for a patent, be it remembered, does not grant the right to *make* or *use* an invention, but only the right to prevent others from doing so. The government has given B a patent on his specific improvement, although it is to be remembered that there has been a prior generic patent issued to A, broadly covering all couplers employing a combination of a head, a knuckle and a locking pin. B's patent, therefore, does not give him the right to make or use his own invention, because his invention cannot be made or used, except in making or using the invention which is already patented by A. Obviously, if this were not so, the value of A's broad patent would be destroyed as soon as anyone patented an improvement upon it.

The enforcement of such a rule would practically upset the whole patent system, since nearly every invention is, or may

be, broad or generic to others, in the same line, which follow after.

What the government does give to B is simply the right to *prohibit anyone else* from using that which he originated, or his particular contribution to the art, which in this case was a specific improvement upon the locking pin or knuckle of A's coupler. A, until his patent expires, can, if he chooses, entirely prevent B from putting his invention in practice, for the reason that B's invention is of such a nature that it cannot be used except in conjunction with the invention made by A. In the words of the patent practitioner, it is but one specific form of a generic structure, of which A holds the monopoly.

On the other hand, while A is entitled to prevent B, as well as everyone else, from making or selling any couplers embodying the broad or generic invention on which A holds a patent, B, by virtue of his patent and rights thereunder, can entirely prevent A from appropriating or making any use of his specific or improved form of knuckle or locking pin. If A wants to incorporate B's improvement in his coupler, he must get the consent of B by license or purchase. In the absence of such consent, he is confined to the use of his generic form of coupler without B's improvements.—"American Machinist."

GALVANIC ACTION OF COPPER SHEATHING.

The opponents of copper sheathing for our warships will find strong confirmation of their criticisms in the very ugly discovery that was made the other day on the British cruiser *Ariadne*, and later on another cruiser, the *Spartiate*. Copper sheathing was introduced to enable ships to remain at sea for longer periods than is possible when the steel plating of the hull is unprotected. The theory is good, and the practice of it would be equally so were it not for the disadvantage that the galvanic action of copper on other metals in the presence of salt water is liable to be very destructive. As long as the metal bolts which

fasten the wood sheathing to the inner steel hull can be kept perfectly watertight, and the salt water prevented from getting in contact with the steel hull, sheathing is no doubt an excellent device; but experience seems to prove that it is extremely difficult to preclude conditions favorable to galvanic action. In some cases the steel plating has been badly attacked, while in others, the fitting of the sea valves and other outboard connections have been seriously damaged. During the latter part of August, the *Ariadne*, a new British cruiser of 11,000 tons displacement and 21 knots speed, was found to be leaking rapidly while at her moorings. The leak was of such dimensions that it was assumed that a Kingston valve had inadvertently been left open.

After the vessel had been hurriedly docked, it was ascertained that the corrosion of the bolts, both inside and outside of the ship, had been so extensive that the mounting of one of the under-water fittings had fallen off, allowing a great inrush of water. The corrosion of the outer bolts of the *Spartiate*, a sister ship, had previously been detected, and on docking the vessel it was found that the inner bolts had also been corroded. Referring to this most serious incident, "The Naval and Military Record" pertinently remarks: "One cannot fail to smile at the comic side of the question, since here we have ships coppersheathed in order that they may keep the sea for lengthened periods, and yet, because they are coppersheathed, they cannot go through the peace maneuvers without becoming so leaky as to threaten their own safety."

We understand that signs of similar corrosion at the same point have appeared in our own sheathed cruiser *New Orleans;* and, in view of these facts, we think that a most searching investigation of the whole subject should be made before we proceed with the proposed copper sheathing of our own cruisers and battleships. Certainly the incident will greatly strengthen the position of the opponents of copper sheathing.—"Scientific American."

TEST OF MANGANESE-BRONZE PROPELLER BLADE AND HUB COUPONS, MONITOR No. 7, THE *ARKANSAS*.

Mark on test piece.	Diameter of test piece.	Sectional area test piece.	Elastic limit of test piece.	Elastic limit per sq. in. sectional area.	Ultimate strength of test piece.	Breaking strength per sq. inch sectional area.	Elongation of test piece in 2 inches.	Per cent. elongation in 2 inches.	Per cent. reduction of area.
	Inches.	*Square inches.*					*Inch.*		
95	1.0	.7854	30,000	38,197	55,300	70,410	.62	29.7	30.4
96	1.0	.7854	29,000	36,924	55,800	71,047	.59	29.5	31.4
97	1.0	.7854	29,500	37,560	56,450	71,874	.55	26.6	26.5
98	1.0	.7854	28,000	35,651	53,330	67,902	.58	28.1	27.7
99	1.0	.7854	29,000	36,924	56,400	71,811	.65	32.5	30.4
100	1.0	.7854	30,000	38,197	53,800	71,047	.60	30.0	31.3
101	1.0	.7854	30,000	38,197	56,100	71,429	.49	24.5	24.8
102	1.0	.7854	29,000	36,924	53,900	68,627	.52	26.0	26.4
Averages, ...			29,313	37,322	55,385	70,518	.58	28.8	28.8

NOTE.—Test pieces 95 and 96 are hub coupons.
Test pieces 97, 98 and 100 are starboard blades' coupons.
Test pieces 99, 101 and 102 are port blades' coupons.

September 12, 1900.

NOTES.

THE RUSTING OF IRON.

Ordinary red rust is essentially an oxide of iron containing combined water, and practically identical with jeweler's rouge, which is nearly pure ferric oxide, Fe_2O_3. The ordinary chemical explanation of the process is that the iron combines with the oxygen of the air according to the equation

$$2Fe_2 + 3O_2 = 2Fe_2O_3,$$

or taking the combined water into account according to the equation

$$Fe_2 + 3O + 3H_2O = Fe_2O_3, 3H_2O.$$

That these simple equations do not represent the actual facts of the case has been known for years, as well as the fact that iron placed in contact with an alkali does not rust. To explain this fact it has been suggested that the presence of carbonic-acid gas is necessary for rusting to take place, and that the alkalies absorbing this prevent rusting. On this view the phenomenon of rusting, according to Professor Crum Brown, takes place in two stages, as follows:

1. $4[Fe + H_2O + CO_2] = 4FeCO_3 + 4H_2$.
2. $4FeCO_3 + CH_2O + O_2 = 2Fe_2(OH)_6 + 4CO_2$.

A close investigation of the matter has, however, led Mr. W. R. Dunston, F.R.S., to question this theory, and to conclude that the presence of CO_2 is by no means necessary for the production of this red rust. Thus five pieces of bright iron were taken, the first was placed in dry oxygen, the second in water vapor, the third in liquid water, the fourth in a mixture of pure oxygen and water and vapor. In none of these cases was any rust formed. The fifth specimen, however, was placed in liquid water charged with pure oxygen, and the production of rust was then quickly apparent, although no CO_2 was present. In fact, for this rust to be formed it is essential, Mr. Dunston finds, that the iron shall be in contact with liquid water and with oxygen.

A piece of bright iron placed in a receiver over a dish of water remained perfectly bright for sixty hours, provided that the temperature was kept constant, so as to prevent the con-

densation of the water on the metal. A similar piece, similarly suspended, but without any precaution against this condensation soon showed spots of rust. Reasoning on the results of the experiments just described, and on the fact that the alkalies prevent the formation of rust, Mr. Dunston suggests that the real active agent is hydrogen peroxide. To test this, iron was suspended in water, to which were added substances, some of which decompose this peroxide, whilst others have no such action. In every case the former prevented the production of rust, whilst it always occurred in the latter. In addition to the alkalies which destroy peroxide, potassium bichromate, potassium chromate, potassium ferrocyanide, and sodium nitrate, were tried. These, whilst destroying the peroxide, are, unlike the alkalies, powerful oxidizing agents, yet nevertheless they proved efficient protectors of the metal. In another experiment a piece of iron was immersed in a solution of hydrogen peroxide, which was made just alkaline. Under these conditions, although the peroxide was decomposed and oxygen was freely liberated at the surface of the metal, the latter did not rust. The whole phenomena of rusting is then, according to Mr. Dunston, effected by the iron taking up O from the water, forming ferrous oxide, and producing peroxide of hydrogen; the latter then immediately combines the ferrous oxide to form the red rust.—"Engineering."

PETROLEUM FUEL IN METALLURGY.

The disadvantages of solid fuel are now very generally recognized, and in another column we have discussed the desirability of using gaseous fuel for metallurgical operations. In many cases, however, it is desirable to use liquid fuel, without the intermediate stage of conversion into gas, and in this connection the paper of M. G. de Krivochapkine in a recent issue of "Le Génie Civil," will be of interest.

In Russia, where petroleum is readily available as a fuel, there has naturally been attempted various methods of utilizing it to the best advantage, especially by the firm of Nobel Brothers, by

whom the petroleum industry has been so extensively developed. As long ago as 1882 M. Nobel presented several communications to the Russian Imperial Technical Society at Moscow, and made the experimental demonstration of the fusion of a ton of cast iron by the combustion of 66 kilogrammes of petroleum residue.

The principal innovation made by Nobel was the special form of grate upon which the petroleum residue was burned. This arrangement consists of a series of superposed troughs, placed at the entrance of the furnace, the liquid combustible being allowed to flow into them under the control of a device by which a constant level is maintained. A space is provided between each trough for the admission of air, and when the fuel is ignited the draft of air over the troughs maintains a complete combustion without requiring the use of steam jets or compressed air, thus giving a marked advantage over similar devices. The stream of burning gases from this device may be employed in a variety of ways. It may be directed over a series of crucibles for melting steel, in which case experiment has shown that 0.6 ton of petroleum residue has sufficed to melt a ton of steel, as compared with the combustion of 1.5 tons of coal.

For the evaporation of water it has been customary to use the steam atomizer, but with the Nobel system it has been shown that 14.35 liters of water have been evaporated per kilogramme of residue, as compared with 11.4 liters with the use of the atomizer. At St. Petersburg and in the Caucasus this arrangement has been used in connection with the reverberatory furnace for the fusion of copper in crucibles. The general arrangement does not differ greatly from the ordinary reverberatory furnace, the petroleum burners being placed in the position of the furnace door, and the grate being covered. The flame from the burning petroleum then streams over the bridge wall upon the hearth, and, with the proper admixture of air, produces an intense heat. The use of petroleum residue is of special importance in this connection in the Caucasus, in which wood and coal are almost entirely lacking, and numerous industries have

grown up in the Baku region, depending upon petroleum residue as fuel.

Another application of petroleum as fuel is found in the so-called Quarnström furnace. This is a form of portable petroleum forge, especially adapted for heating rivets. In this furnace the troughs containing the petroleum are placed at the bottom, with air spaces between, supplied by the natural draft. This furnace appears to work very well, but in order to avoid the production of too much soot it has been found best to employ the lighter hydrocarbons, instead of the petroleum residue.

An important application of the petroleum furnace has been in connection with glass works, in which the Kroupsky regenerator has been employed to advantage. In this case the furnace is provided with regenerators similar to the Siemens regenerative furnace for gaseous fuel, but instead of using the chambers of highly-heated brick checker work to heat the incoming air, they are employed to gasify the petroleum vapor, and thus render its combustion more complete and intense.

In connection with the manufacture of glass the Malichef furnace has also been employed, this more nearly resembling the ordinary Siemens regenerative furnace. Here the vapor of petroleum and the air both pass through the regenerative chambers, and are thus prepared for most complete combustion. It is asserted that in these furnaces 1.3 tons of petroleum residue will melt 4 tons of glass.

These illustrations show that Russia is making active progress in the utilization of petroleum as fuel, and that possibly other parts of the world may have some lessons to learn from her.

THE BÁNKI MOTOR.

An interesting feature of modern mechanical development is the marked attention which is being given to heat motors other than the steam engine, and the remarkable development of various forms of the internal-combustion motors is most interesting. Among the more recent inventions in this line of work we

have already noticed the motor designed by Professor Donát Bánki, of Budapest, and now we have very full accounts of its latest performances in recent issues of the "Zeitschrift des Vereines deutscher Ingenieure" and in the "Zeitschrift des Oesterreichen Ingenieuer und Architekten Vereines," from which we make some abstracts.

It is now generally realized that in order to obtain high economy in an internal-combustion motor a high compression must be employed. In the ordinary gas or petroleum motor the limit to the compression is attained when the temperature of compression reaches the ignition temperature of the mixture of gas or vapor and air. If this ignition temperature is reached before the end of the compression stroke there will be a premature explosion, the motion will be reversed and the motor will fail to operate satisfactorily.

In the case of the Diesel motor this difficulty is avoided by compressing the air alone, and only injecting the liquid fuel at the moment when the combustion is desired, and this method also utilizes the heat of compression as an igniting device.

Professor Bánki adopts another method. The charge of mixed air and hydrocarbon vapor is drawn into the cylinder as usual, but with it is also mingled a fine spray of water. When the high compression is given to this mixture the water absorbs a portion of the heat of compression and thus the temperature may be kept below the ignition point. When the combustion does take place the conversion of the water into steam adds to the expansive power of the charge and thus aids in the power stroke, and at the same time the absorption of heat in the generation of this steam aids in keeping the cylinder from becoming overheated.

In the paper of Professor Meyer, above referred to as appearing in the "Zeitschrift des Vereines deutscher Ingenieure," the construction of the motor is clearly shown by drawings and sections, and in general it differs but little from an ordinary vertical gas or petroleum motor. The principal addition lies in the device attached to the inlet passage to provide for the admixture of the water spray.

With this arrangement of parts it has been found possible to work with as high a compression as 16.4 kilogrammes per square centimeter (233 pounds per square inch), without causing pre-ignition of the mixed charge, while the pressure of explosion attained is 39 kilogrammes (540 pounds).

In the tests made by Professor Meyer and others, the fuel used was benzine, this being found most satisfactory, and very full data and results are given for a series of trials upon a motor of 20 horsepower. The motor was operated at powers ranging from 26 down to 8 brake horsepower, and showed a thermal efficiency, on the brake, of 28 per cent when operating at full load.

When it is understood that this includes the internal resistance of the motor, and that trials showed the mechanical efficiency of the machine to be only 74 per cent., it will be seen that this is an excellent performance, decidedly higher than would have been possible under ordinary conditions. The action of the injected water is well seen in the diminished proportion of heat rejected in the cooling water of the cylinder jacket. Instead of 40 per cent. of the heat being carried off in the cooling water, as is the case with gas engines of the ordinary construction, the loss in this way ranged from 21 to 27 per cent., thus showing that a large proportion of the heat usually rejected in the jacket was taken up by the water into the cylinder, where it was converted into work.

The idea of injecting water into the cylinder of an internal-combustion motor is not entirely new, and Professor Meyer points out the fact that in the early gas engine of Hugon, water injection was used to cool the cylinder. In such an engine, however, there was little or no compression, and the injected water did not act to improve the efficiency of the machine, while in the Bânki motor the action has been combined with the use of high compression in such a manner as to constitute a material improvement.

It remains to be seen to what extent this form of motor can be satisfactorily used with petroleum and the heavier hydrocarbons, since it is possible that the cooling effect of the water

spray may limit the machine to the use of benzine and similar fuels.

In any case it is an interesting example of the ingenuity which is being applied to the development of the internal-combustion motor, which will doubtless ultimately aid in establishing its superiority as the heat motor of the immediate future.

QUICKSILVER.

Some concern is felt in trade circles on the subject of quicksilver, and the fear that the world's supply is proving unequal to the growing demand. There is no actual famine, but stocks are not considerable, and, so far as can be ascertained, they are decreasing; while at the same time, the price keeps up at a high level not attained for several years previously. There are no complete and authoritive statistics of the annual output for the whole world. Some totals come along now and again on French journalistic authority, but they are ludicrously inaccurate and worse than useless. Messrs. Wm. Sargant and Co. furnish some figures in their annual metal circular, showing the quantity obtained from the principal sources, and they are useful in appraising the situation, especially with the aid of the Board of Trade monthly returns. The production of Spain, Italy, Austria and California in 1899 was 94,935 bottles, a total exceeded in six previous years since 1890, the exception being 1897, with 91,906 bottles; 1896, with 89,070 bottles; and 1892, with 94,640 bottles. The appended Table gives details:

	1899.	1898.	1894.	1890.
	Bottles.	*Bottles.*	*Bottles.*	*Bottles.*
Imports of Spanish into London	45,729	46,367	42,414	50,202
Imports of Italian*	6,206	5,650	8,700	12,470
Austrian production	15,000	15,000	15,000	14,000
Californian production	28,000	28,000	30,183	22,926
Total of principal sources	94,935	95,017	96,297	99,598
Exports from United Kingdom*	33,101	32,288	41,046	56,702
	£. s. d.	*£. s. d.*	*£. s. d.*	*£. s. d.*
Highest price of Spanish	9 12 6	7 15 0	6 15 0	10 7 6
Lowest price of Spanish	7 15 0	7 0 0	5 10 0	8 17 6

*Calculated November to November.

The price of quicksilver this year has been steady in the neighborhood of £9 12s., or nearly up to last year's highest, and it is to be observed, as showing how quicksilver has sympathized with the generally high range of values, that only in 1890 (among recent years) have present prices been surpassed, £7 representing the annual average for the intervening years. There has been no striking decline in supplies in the past decade, but the trouble is that there has been no expansion to meet the growing demand, more particularly in connection with gold-mining. If we go further back, we get a better notion of the decrease. The production from the cinnabar mines of California, the only large mines on the American continent, with one insignificant exception, the Texas mine, has decreased from 79,936 bottles in 1888, to 28,100 bottles last year. The Almaden in Spain and the Idria in Austria, both of which have been worked and have produced for nearly 2,000 years, and the mines of Aurbach & Co., in Russia, are the only other producers of note in the world, though there are being worked a few scattered mines of minor importance in Spain, Italy, Siberia and elsewhere. The total Spanish output last year is estimated by the "Revista Minera" at 39,339 bottles, apportioned thus: Almaden mines, 37,378 bottles; El Porvenir Asturias, 1,475 bottles; La Union Miere, 300 bottles; La Soterrana Asturias, 61 bottles; various mines in Granada, 125 bottles. The Almaden mines raised 20,322 tons of ore, and smelted 15,194 tons.

If these figures of output are to be accepted as accurate, a considerable quantity was taken out of stock for export to the United Kingdom. About 8,700 bottles are obtained in the Mikitowka mines in Russia, but they do not come to the English market, which is supplied almost solely from Spain and Italy, the last, however, being of relatively little account. The Italian mines are in the neighborhood of Monte Amiato, which now produce 200 tons or so in a year, against 274 tons in 1893. The greater part of the Californian quicksilver goes to Mexico, to be used in mining operations there. A certain proportion is consumed by mines in some of the Western States, and the rest finds markets in various places all over the country. A part

from its use for gold and silver mining, to aid in the extraction of these metals, the chief demand for mercury is as a drug. Its chlorides—corrosive, sublimate, red oxide and calomel—are well known. From the cinnabar, which is its most abundant ore, vermillion is prepared.

Very little enterprize has been displayed in recent years in the exploitation of quicksilver deposits, but it is clear that there are openings. As a result, some attention is coming to be paid to those mines which are not yet as "well held" as are the Almaden and Californian properties, but which afford promise of a more than respectable profit. This is especially the case in the colonies. Quicksilver appears to be well distributed in New South Wales. It has been found in the Cadgegong river; at Bingara, where there are several diamond mines, in the vicinity of the Solferino goldfields; and at Cooma, at the entrance to the New South Wales snow country, where the assays of ore yielded 25 per cent. of quicksilver. The richest deposits have, however, been discovered near Yulgilbar, in the Clarence River district, where considerable developments have been made, and six distinct shafts put down on three parallel lodes.

Several tons of the ore from these lodes have been brought to Sydney, and have averaged from 3 to 5 per cent. of mercury, and it has been ascertained that the "spent" ore contains gold and silver. The area of the ground examined by the Government geologist is about 120 acres; but it is believed that, with the progress of prospecting operations, other lodes, more or less rich, will be found. A further examination of the locality is being made by the Government geologist, and his report is about due. He has definitely ascertained the existence of three distinct parallel lodes, improving as they go down. Machinery is being erected, and a preliminary testing of about 1,000 tons of ore will be made. In various parts of New Zealand, again, quicksilver, either free or associated with sulphur as cinnabar, has been found from time to time. At Tokomairo and Russell native quicksilver is found; and at Puhipuhi, Ohaawai, Kerikeri, and the Bay of Islands generally in the North Island and at Waipori, in the Province of Otago in the South Island, quick-

silver is found as cinnabar, mostly in the shape of rolled fragments of pure sulphide. The Ohaawai quicksilver deposits are the most important yet discovered, and although their existence was known as long ago as 1870, when Captain Hutton inspected them, it was not until a few years ago that they came under the notice of English capitalists. They have been reported upon favorably and appear to have a respectable future.—"Engineering."

SPONTANEOUS COMBUSTION OF COAL.

In view of the large export trade in American coal, a trade that promises to become much larger, the report of an investigation recently made for the London Board of Trade is of exceptional interest. This report was made by Captain Castle and Professor Threlfall, F.R.S., into the circumstances attending the serious damage, through fire, sustained by the sailing ship Walter H. Wilson, while carrying a cargo of Cardiff coal. The investigation was ordered by the Board of Trade because of the numerous cases of mysterious fires in coal cargoes. The report of the investigation is long, and amounts to a treatise on the spontaneous combustion of coal, to which the fire in the vessel was attributed. It suggests the possible cause of the disappearance of many vessels carrying coal from New South Wales.

The conclusion of the investigators is that coal is a dangerous cargo when stowed as the custom is at present. The finding is that coal can be stacked to a given height with perfect safety, and this height is 17 feet. When this height is exceeded, spontaneous combustion sets in.

Another fact was ascertained which is surprising, viz: That wet coal is less liable to heating and spontaneous combustion than dry coal. Heretofore the reverse was supposed to be true. This fact was ascertained by actual experiments made with two bins, each holding 245 tons of coal—the coal in one bin kept wet, the other perfectly dry. In the dry bin the temperature rose in 60 days to 135 degrees, while in the bin that was kept wet the temperature remained at about 40 degrees.

We summarize Prof. Threlfall's conclusions, which will be

interesting and, perhaps, valuable to American shippers: "The way to prevent spontaneous fires is to wet the coal. In the case of cargoes of large coal, the only danger is in the heap of small coal under the hatchways. If the coal under the hatchways is thoroughly wetted, the danger of spontaneous combustion will be avoided. In the case of small coal, the whole mass should be thoroughly wetted either before or during loading. The coal may be wetted either by salt or fresh water."—"The Marine Journal."

MECHANICAL HEATING AND VENTILATION.

The blower system of mechanical ventilation and heating, which is being used so extensively for heating and ventilating machine shops, as well as other buildings, has advantages that seem to commend it to an increasing number of users. The particular features of this system are summarized in a concise manner by Walter B. Snow, of the engineering staff of the B. F. Sturtevant Co., as follows:

"The entire heating surface is centrally located, enclosed in a fire-proof casing, and placed under the control of a single individual, thereby avoiding the possibility of damage by leakage or freezing incident to a scattered system of steam piping and radiators. The heater itself is adapted for the use of either live or exhaust steam, and provision is made for utilizing the exhaust of the fan engine, thereby reducing the cost of operation to practically nothing. At all times ample and positive ventilation may be provided with air tempered to the desired degree. Absolute control may be had over the quality and quantity of air supplied. It may be filtered and cleansed, heated or cooled, dried or moistened at will. By means of the hot and cold system, the temperature of the air admitted to any given space may be instantly and radically changed without the employment of supplementary heating surface.

The pressure created within the building is sufficient to cause all leakage to be outward, preventing cold inward drafts and avoiding the possibility of drawing air from any polluting source within the building itself. By returning the air, using live

steam in the heater and operating the fan at maximum speed, a building may be heated up with great rapidity, as is usually desirable in the morning.

The area of heating surface is only one-third to one-fifth that required with direct radiation, while the primary cost and operating expense of a fan is far less than that of any other device for moving the same amount of air.—" Machinery."

PULVERIZED COAL AS FUEL.

Some peculiarly novel and interesting experiments made recently in Chicago with pulverized coal as fuel indicates the possibility of a radical change in the coal-using conditions of great manufacturing plants requiring steam power.

Even with all the best modern appliances at command, the burning of ordinary coal in furnaces is a process so wasteful when results are compared with the potential energy liberated, that a veritable flood of inventive talent has been poured around the problem of increased fuel efficiency. Pulverized coal is a contribution to the list of attempts at solution which have already shown valuable results under careful tests, and may eventually supersede the familiar form of this fuel. As shown in the experiments at Chicago, conducted under the direction of Illinois Central railway officials, the new preparation of coal burns in an ordinary furnace just like gas, with no waste and no residuum. The finely pulverized coal, blown into the furnace with a due admixture of air, flashes into gas and is instantly consumed. An increase of from 25 to 40 per cent. in the steam producing capacity of the coal used has been certified to by the engineer experts who conducted the Illinois Central series of experiments.

Manifestly, if the results thus shown can be reproduced at will elsewhere, there will be little delay in the general installation of the new process of fuel treatment and combustion. The saving in coal alone in one year would amply cover the cost of the new machinery required, while the further relief from noxious gasses, slag, cinders and waste, and from the intolerable

nuisance of heavy black smoke, would be a public boon of great value. It is to be trusted that this elaborately heralded new device is not, like so many others, a mere pretense and illusion destined to early obscurity and oblivion.—" Marine Record."

PETROLEUM INDUSTRY IN RUSSIA

The St. Petersburg " Messenger of Finance" states that English capitalists have recently invested $9,000,000 in the petroleum-producing territory between the Caspian and Black Seas.

Since 1892, when new regulations concerning the petroleum industry were published, more than thirty firms have received permits from the Minister of Agriculture and Crown Domains to engage in the petroleum industry in Russia, and applications are being made for permission to develop the territory on the Apsheron Peninsula, some 300 miles north of Baku, on the Caspian Sea.

This increased demand for fuel caused foreign capitalists to look carefully after the residues, which has resulted in a decrease in the amount of kerosene exported and a greater demand for petroleum fuel at good prices; this increased 900 per cent. during the past 6 years and is still growing, notwithstanding the competition of pit coal and wood. The demand for petroleum fuel from the factories in and about Moscow has more than doubled in six years and now amounts to millions of tons annually.

The consumption of liquid fuel on locomotives has increased tenfold in 9 years and on river steamboats fourfold in 11 years, while the consumption of coal increased 113 per cent. and of wood 58 per cent.

In order to meet the demand for liquid fuel, Baku must increase her output from 242,000,000 poods to 360,000,000 poods (3,903,226 tons to 5,806,451 tons), and, in order to supply the increasing demand from the interior of Russia, it will be necessary to increase the output of residues to 8,064,516 tons in the near future. This development will require a further investment of $8,000,000, which must come from foreigners.

The slow development of the export of kerosene is due partly to the fact that Baku finds it more profitable to produce liquid fuel and partly to the competition of the Standard Oil Co., but, unless the export of Russian kerosene is increased, as the demand from the interior is small, the liquid fuel will contain too great a percentage of dangerous volatile products for profitable trade.

The increased production of kerosene is necessary to supply the fuel residues, and, while the same will meet strong opposition from the United States, success is only possible by Russia adopting the Standard Oil Co.'s methods for saving, transporting and delivering the product, which will require a largely increased capital. It is estimated that the amount of $10,000,000 will be required to successfully compete with the American oil in England alone.—" Marine Record."

AMERICAN AND BRITISH BLAST-FURNACE PRACTICE.

At a time when American products of iron and steel are crowding British products in home and colonial markets, the paper of Mr. T. B. Rogerson, before the West of Scotland Iron and Steel Institution, is both appropriate and interesting.

In making the comparison, Mr. Rogerson is in a position to speak whereof he knows, since he has spent about eight months recently in the United States, visiting most of the large blast-furnace plants in the iron-making States, and being given excellent opportunity to observe American practice.

The high reputation of Scotch pig iron renders it especially appropriate that the Scotch furnaces should be selected by Mr. Rogerson for comparison with the American ones; and, in fact, with the exception of the fact that English furnaces use coke and the Scotch use raw coal, the Scotch practice is fairly entitled to be considered representative British practice.

The Scotch furnaces are about 65 feet high and 15 to 17 feet diameter of bosh, and are worked at a blast pressure of from 5 to 6 pounds, the blast being heated to 1,100 degrees Fahrenheit to 1,300 degrees Fahrenheit in Cowper or Whitwell stoves, the

hot-blast main being usually common to all the furnaces in a plant. The coal is handled in charging barrows, at the rate of one ton per barrow, being taken up a steam hoist to the top of furnace and dumped in by means of the ordinary bell and hopper arrangement. This is followed by the barrows containing the mixture of ore and limestone, all being hand-filled, the men doing about 60 tons per day each. The slag is taken away from these furnaces either by running it into holes in the slag fall and hauling the pieces away when set, or else the liquid slag is hauled away in tubs or ladles of 4 to 12 tons' capacity. The iron is run out of a tapping hole into a sand bed and cast into pigs, and when cold is carried away by hand, broken, and deposited in trucks for conveyance to the steel works. The tuyeres are cooled by being made of coiled pipe, through which water flows, while the slag tuyere is formed of two annular brass cones, with water space between. The blast tuyere is stopped with specially-prepared clay, and the furnace must be thrown off blast to do this, but the brass tuyere is stopped with an iron plug, which can be done with the furnace on blast. The tapping hole is stopped by hand with specially-prepared clay, and it generally takes about ten minutes to do this, the furnace being off blast during this time.

English blast-furnace practice is very similar to Scottish, except that coke is almost entirely the fuel, the labor, or form of filling, etc., being the same as in Scotland. In a few places, however, they use blast up to 8 and 10 pounds pressure, and have also apparatus for lifting the pig iron, either in the whole grid, and breaking it with a specially-constructed machine, either hydraulic or steam, and delivering it sampled into railway wagons, or gathering it up into slings of about 2 tons, and delivering into wagons.

With these few exceptions there are no labor-saving devices other than those which were in force thirty years ago in Great Britain. The maximum output of a British furnace is about 1,200 tons per week, with 10 pounds blast pressure.

In the United States the furnaces are much larger, the output per furnace greater, and the cost of production lower than in

Great Britain. The furnaces are from 80 to 100 feet high, and from 17 to 20 feet bosh, with 8 to 16 tuyeres. Hot-blast stoves are used, but each furnace has its own outfit, so that the blast can be varied in each case as desired. The blowing engines are more powerful than in England, furnishing blast at a pressure of 10 to 18 pounds per square inch, and the more recent engines press up to 25 pounds, a single engine making blast sufficient to make 400 tons of pig iron per day.

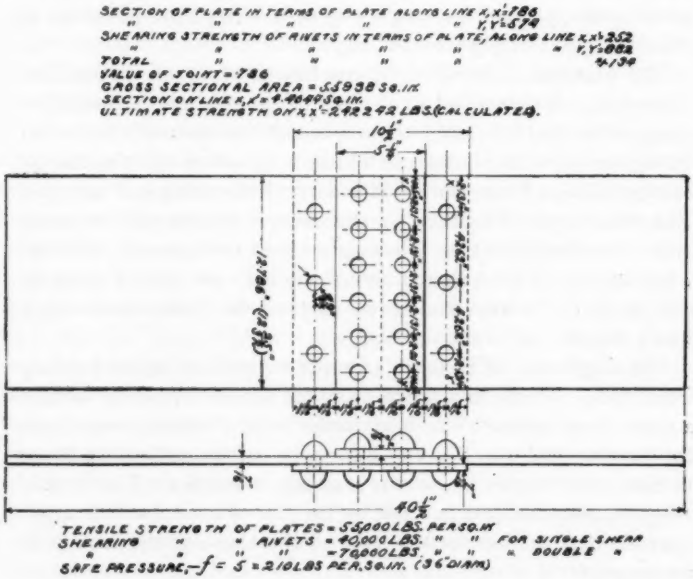
The material is handled by machinery as much as possible. The charge is conveyed to the top of the furnace in specially-designed buckets, holding about 4 tons, the bottom of the bucket fitting the top of the furnace bell in such a manner that the charge can be delivered into the furnace without the escape of any gas. The time required to take up the charge, deposit the contents, and bring the bucket back, requires about two minutes, and the whole filling of the furnace, about 700 tons per day, is done by one hoist engineman and three men on the locomotive which hauls the charge to the furnace.

The slag is run off through a copper tuyere into ladles holding from 10 to 15 tons, and taken away for deposit in liquid form in a convenient place. The iron comes in the ordinary way from the tapping hole, and is run into large ladles, and taken direct to the steel works or to the casting machines. The hole is stopped, after the cast is done, by the use of a mechanical appliance, so that neither in the case of the slag nor the iron is it necessary to shut off the blast.

The great makes of the American furnaces are due to the greater pressure of the blast, bringing the charge down very rapidly and causing quick smelting even with the large proportion of small rich ores used. When the output of the furnace is raised to 2,000 to 5,000 tons per week the questions of regular filling and the disposal of iron and slag present themselves. The result has been the replacement of the laborious work of men by machinery, and such a cheapening of cost that, in connection with abundant ore and fuel, has enabled the American iron-making industry to reach its present commanding position.—"Engineering Magazine."

TEST OF RIVETED JOINTS IN THIN PLATES.

The following is an extract from tests made at the Watertown Arsenal with riveted joints on $\frac{7}{16}$-inch plates, comparing the strength of these joints when the holes were simply punched, punched and reamed, and simply drilled:


SECTION OF PLATE IN TERMS OF PLATE ALONG LINE x, x' = .786
" " " " " " Y, Y' = .574
SHEARING STRENGTH OF RIVETS IN TERMS OF PLATE, ALONG LINE x, x' = .252
" " " " " " Y, Y' = .082
TOTAL " " " " " .334
VALUE OF JOINT = .786
GROSS SECTIONAL AREA = 5.5938 SQ. IN.
SECTION ON LINE x, x' = 4.40955 SQ. IN.
ULTIMATE STRENGTH ON x, x' = 242292 LBS. (CALCULATED).
10 1/2"
5 1/4"
18 3/16"
(18 3/16 - 10 3/8) = 7 13/16"
40 1/2"
TENSILE STRENGTH OF PLATES = 55,000 LBS. PER SQ. IN.
SHEARING " " RIVETS = 40,000 LBS. " " FOR SINGLE SHEAR
" " " " = 70,000 LBS. " " " DOUBLE "
SAFE PRESSURE, f = 5 = 2101 LBS PER SQ. IN. (36" DIAM)

RESULTS.

	Breaking loads in pounds.		
	Holes punched.	Punched and reamed.	Drilled.
Total for piece, ...	261,600	286,800	308,200
Per square inch, full section, . .	43,890	48,780	50,030
Per square inch reduced section,	53,672	61,778	63,492

NEW SHIPBUILDING PLANT.

The new shipbuilding plant of the Risdon Iron Works, of San Francisco, Cal., is rapidly assuming material form. This company has purchased for its new plant the property of the Pacific Rolling Mill Co. This property consists of some 32 acres of ground immediately adjoining the Union Iron Works, and has a deep-water frontage of over 1,700 feet. Application has been made to the Board of State Harbor Commissioners for a site for a new dry dock. This dry dock will be 500 feet in length and 100 feet in width. New wharves are now in course of construction. Several new steel-frame buildings are contemplated, and large orders covering the structural materials for the following buildings have recently been placed: Machine shop, length 308 feet, width 100 feet, total lift over main floor 45 feet; boiler shop, length 140 feet, width 180 feet, total lift over main floor 40 feet; machine and ship blacksmith shop, length 300 feet, width 60 feet; foundry, length 177 feet, width 160 feet; ship joiner shop, length 140 feet, width 81 feet.

The machine shop will be equipped with four electric traveling cranes with capacities of 50 tons, 20 tons, 15 tons and 10 tons. Each crane will be of the most modern type and equipped with 5-ton auxiliary hoists. The boiler-shop crane equipment will consist of one 60-ton, one 20-ton and one 10-ton electric traveling crane, each equipped with 5-ton auxiliary hoists. The foundry crane equipment will consist of one 30-ton and one 20-ton electric traveling crane with 5-ton auxiliary hoists. The ship blacksmith shop will be equipped with eight jib cranes of 5 and 10 tons capacity each. The machinery equipment throughout will consist only of the most modern tools. The entire plant, with the exception of the steam hammers in the blacksmith shop, will be operated by electricity, and contracts have recently been placed with the Independent Electric Light & Power Co., of San Francisco, to furnish power for this plant. Direct current will be used for the cranes and an alternating current for the machinery equipment. Separate motors will be installed in each department. The cost of electric power for operation of this plant for the first year will amount to over $30,000.—"Engineering News."

ABOUT ALUMINUM.

Aluminum is cheaper than all the common metals excepting zinc, lead and iron. Brass, tin, copper, are of approximately the same specific gravity, and in comparing their costs we usually think only of their relative cost per pound; but aluminum is entirely in another class. It takes only one-third of a pound of aluminum to take the place of one pound of these metals, and the proper basis of comparison here is to compare the price of one-third of a pound of aluminum with that of one pound of brass, etc. The comparison, therefore, stands as follows:

One-third pound of aluminum, ...	$0.11
One pound of brass, ...	0.15
One pound of copper, ...	0.17
One pound of tin, ...	0.30

The next great improvement, after that of the fall in price, has been in the successful manufacture of light, strong alloys. This has been a subject at which metallurgists have worked hard and long, and their labors are bearing fruit in abundance. Pure aluminum has many resemblances to pure copper. Take away the red color of copper, and its softness, malleability, toughness, silky-fibrous fracture are almost exactly duplicated by aluminum; but they are both soft, rather weak metals. Five per cent. of silicon or manganese, or 30 per cent. of zinc, added to aluminum, make strong metals as rigid as bronze yet only one-third as heavy.

Such light, strong, good casting and machining alloys have a large field of usefulness. One, for example, made by the Delaware Metal Refinery, of Philadelphia, is a hard white alloy, specific gravity 3.1, melts clean, runs fluid, makes beautifully sharp and perfect castings, turns and machines like the finest brass, polishes well, and, to conclude, is fully as rigid and strong as gun metal or the best of the ordinary bronzes. This alloy is principally of aluminum and zinc, and sells at the same price per pound as pure aluminum. The field of application of light alloys with such properties to light-running machinery, portable appa-

ratus, vehicles, instruments, etc., is extensive, and the next few years will see its use very general for such and similar purposes.

Electric conductors can now be laid more cheaply in aluminum than in copper. Pure aluminum has over 60 per cent. the conductivity of pure copper, and is fully as strong and resistant to atmospheric influences. It is, therefore, only necessary to take an aluminum wire one-fourth as large again in diameter as a copper wire (giving a little over 50 per cent. more section) to get equal conductivity. Such a wire weighs one-half as much as the copper wire it replaces, and costs only two-thirds as much. Over 500 tons of aluminum were used for trolley-line feed wires last year, and probably double as much will be used this year. Fifteen years ago 2½ tons of aluminum was the output of the whole world for one year.

Aluminum can be rolled out to $\frac{1}{2000}$ of an inch in thickness, and then beaten out to $\frac{1}{40000}$ or even $\frac{1}{70000}$ of an inch. As thin sheet it has found some application in place of cardboard, for business cards, etc., but as leaf it has entirely superseded silver-leaf in decorating. This leaf can, moreover, be ground to powder, and in this condition is used by printers for silvery printing, and as a paint. For the latter use it is simply mixed with a varnish, like ordinary bronzing powders, and has already proven its beauty and utility on Uncle Sam's letter-boxes.—"Journal of Franklin Institute."

MISCELLANEOUS.

THE 600-H.P. GAS ENGINE at the works of the John Cockerill Co., Seraing, Belgium, has recently been tested with excellent results. This is a four-cycle engine, supplied with gas directly from the blast furnaces in the Bessemer mill. The gas passes through dust chambers and then into a cast-iron box, where it is mixed with steam, in order to cool it and increase its density. Calculations showed that the indicated thermal efficiency of the engine rose as high as 27.16 per cent., and that the net efficiency varied between 19.86 and 22 per cent. The mechanical efficiency of the machine varied between 73 and 81 per cent. On an average of the total heat supplied 20 per cent. was converted into work, 52 per cent. passed away in circulating water and 20 per

cent. in the exhaust. The consumption of gas was about 31 cubic meters per H.P. hour. The engine is used to supply blast to the furnaces.—"Engineering News."

ZINC PRODUCTION IN RUSSIA.—The production of zinc in Russia has hitherto been confined to Poland, where this industry has existed since the year 1816. The Poland zinc industry is at present in a very fair state of development, the total for 1898 being 5,500 tons, against 3,900 tons in 1875 and 410 tons in the year 1816. The total production of zinc in Poland, since this industry commenced, amounts to an aggregate of 240,000 tons.—"Engineering."

A PATENT (651,356) has recently been granted on a die for forming twist drills from cylindrical blanks. The principle involved is analogous to that of broaching, the blank being forced longitudinally through the die, which has inwardly projecting cutters of the proper shape for forming the grooves. As the blank is forced through, it is at the same time rotated sufficiently to give the flutes the required twist.—"Machinery."

THE PASSIVE state of iron, in which it resists solution by acids, has long been known, and is now attributed to the formation on the surface of the metal of a thin layer of oxide. The thickness of the layer necessary is extremely minute, since Mr. Oberbeck has shown that in the case of platinum a film of another metal deposited on the surface and only $\frac{1}{1000000}$ millimeter thick suffices to alter the electromotive force of the metal if forming an element of a voltaic couple. In a communication to the "Archives des Sciences Physiques et Naturelle" (Geneva), Mr. F. J. Micheli describes a measurement he has recently made of the actual thickness of this film of oxide. The method followed was to construct a small mirror of iron, and to measure the polarizing angle: first, when the iron was in the active state; and, secondly, when it has been rendered passive by immersion in fuming nitric acid. The difference in the constants of polarization was found to correspond to a film of oxide equal in thickness to one-thirtieth of the wave length of sodium light. The formation of this film had nevertheless sufficed to render the iron electro negative to copper. Similar experiments

made on pure chromium obtained from Dr. Goldschmidt at Essen showed that with this metal the production of the passive state is not due to the surface oxidation of the metal, since the polarization constants of a chromium mirror were identical, whether the metal was in the active or the passive state. In the former state the electromotive force of chromium-platinum couple was found to be 1.58 up to 1.607 volts, and in the passive state 0.443 volts.—" Engineering."

PETROLEUM INDUSTRY IN JAPAN.—Vice-Consul-General McLean, at Yokohama, has sent to the State Department information concerning the rapidly developing petroleum industry in the district of Echigo, Japan, showing that there are at present over 30 petroleum companies in the oil district in Echigo. Some of these possess a capital of over 1,000,000 yen ($498,000), the aggregate capital amounting to 12,000,000 yen ($5,966,000). The proposed scheme to construct a pipe line from the seat of the petroleum wells in Echigo to Tokyo, for the rapid and convenient transportation of kerosense, is now assuming practical shape. The promoters of the enterprize have decided to entrust the Nakai engineering office with the business of making a preliminary survey for the projected work. Mr. Miyagi, graduate of the engineering college of the Tokyo Imperial University, will act as advisory engineer in the survey, which is to be begun at no distant date.—" Iron Age."

CHINESE IRON seems likely to become a serious rival in Japan to the English and American pig iron. It is at present sold in Japan at about £1 per ton cheaper than that which is imported from this country. Iron ore from China is also to be largely used in the workshops of the Government iron foundry at Yawata Mura, Fukuoka Prefecture, Kiushu, which is about to commence work. From June or July next the manufacture of small rails and sheet iron is to begin; but the foundry will not be in a position to furnish rails for railway use till the end of the year. The supply of iron ore will in the main be obtained from mines which are the property of the Government and from Niigata and Iwate Prefectures.—" Engineers' Gazette."

SHIPS.

UNITED STATES.

St. Louis, Milwaukee and **Charleston.**—The following extracts are from the "Circular" of September 20, 1900, issued by the Department, for the benefit of bidders for the construction of these three protected cruisers (authorized by Act of June 7, 1900), and defining the chief characteristics of the ships.

Bids will be invited for—

Class I.—Department's design.

Class II.—Bidder's design.

In order that ample time may be given to all ship and marine-engine builders who may desire to submit their own plans of the hull and machinery, or of either, this circular, defining the general requirements of the Department for the vessels referred to herein, is issued. The general plans may be examined in the bureaus on and after December 1, 1900. The plans will be ready for distribution among prospective bidders on application January 2, 1901, and the bids will be opened at the Department on February 1, 1901.

* * * * *

The Department will reserve the right to reject any and all bids not considered advantageous to the Government, and no bids will be considered which propose to furnish vessels of less than 9,700 tons trial displacement, or of less than 22 knots speed on trial, or having a bunker capacity of less than 1,500 tons of coal.

Trial displacement is to be understood as meaning the vessel and her equipment complete, with 650 tons of coal, plus two-thirds of ammunition, plus two-thirds of total stores and provisions, crew, and 50 tons of fresh water for steaming purposes.

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The maximum time allowed for completion will be limited by the Department to thirty-six months for each vessel, and no

bids will be considered which propose to exceed that limit of time. Failure to complete the vessel or vessels within the time specified in the contracts will involve penalties as follows, viz: $300 per day for the first month succeeding the expiration of the period fixed by the contract, and $600 per day thereafter until the vessel is completed and delivered as prescribed by the contract.

The speed trial of the proposed vessels will be conducted after the vessels are fully completed, in accordance with the terms of the contracts, subject to conditions prescribed by the Department, and if on trial the average speed shall equal or exceed a speed at sea of 22 knots per hour for four consecutive hours with an average air pressure of not more than 1 inch of water in the ash pits the vessels will be accepted, in so far as the speed is concerned.

If the speed falls below 22 knots and exceeds 21 knots an hour the vessel will be accepted, so far as speed is concerned, at a reduced price, the reduction being at the rate of $50,000 a quarter knot deficiency of speed from 22 to 21½ knots, and at the rate of $100,000 a quarter knot deficiency of speed from 21½ to 21 knots.

If the speed falls below 21 knots per hour the vessel will, in the discretion of the Secretary of the Navy, be rejected, or accepted at a reduced price to be agreed upon between the Secretary and the contractor. In case of rejection any money that may have been paid the contractor on account shall be refunded by him.

The final trial of the vessels will not take place until a period of six months has elapsed from the date of preliminary acceptance.

The following elements and qualities are obtained and provided for in the Department's design, and their equivalent must be obtained and provided for in all designs submitted:

The vessels are to be driven by twin screws.

The hull is to be of steel, having a double bottom and close water tight subdivisions.

The arrangement of decks above water is to be such as will

provide ample freeboard and good berthing accommodations, the after body of the vessel being carried to the same height as the forward body.

There will be two steel military masts, fitted with upper and lower fighting tops, with search-light platforms, and with signal yards, and gaffs.

It is contemplated to supply the following boats, viz:

1 36-foot steam cutter,	2 30-foot whaleboats,
1 33-foot steam cutter,	1 30-foot gig whaleboat,
1 36-foot launch,	2 20-foot dinghies,
1 33-foot launch,	1 16-foot dinghy,
2 30-foot cutters,	2 12-foot punts,
1 30-foot barge,	2 Carley life rafts.
3 26-foot cutters,	

* * * * *

The armored protection will be as follows:

a. A partial water-line belt about 200 feet in length, abreast the engines and boilers, said belt being 7 feet 6 inches in width and 4 inches in thickness.

b. Above the main belt, the sides of the ship will be protected for a distance of about 133 feet in length by vertical armor of about 7 feet in width and of 4 inches in thickness, the forward and after ends being connected by transverse armor of 3 inches in thickness.

c. Above the lower casemate the sides of the ship will be protected for a distance of about 133 feet in length by vertical armor about 7 feet 2 inches in width and of 4 inches in thickness, with transverse armor 3 inches thick connecting the ends and forming a central casemate with four armored sponsons.

d. On the main deck the four 6-inch guns at the corners of the superstructure will be protected by vertical armor of 4 inches in thickness and about 7 feet 6 inches in height in the form of sponsons.

e. There will be an oval conning tower of 6 feet and 8 feet inside diameters, and 8 feet in height with a shield, both of 5 inches in thickness, the top to be 3 inches in thickness.

f. The conning tower will have an armored tube of 3 inches in thickness, extending to the protective deck.

g. There will be a signal tower 8 feet 6 inches in height, 6 feet in diameter, and of 4 inches in thickness.

The main armor belt is to be so placed that the upper edge will be at least 15 inches above the water line at maximum deep-load draught.

The 6-inch guns on both decks will be further protected by splinter bulkheads of 2-inch nickel-steel placed between each pair of guns on the gun deck, and as shown on the plans on the main deck.

* * * * *

A proctective deck of nickel-steel is to extend throughout the vessel, and will be 2 inches thick on the flat and 2½ inches thick on the slopes. On the slopes this deck will be laid on ½-inch steel plates.

There will be a cellulose belt, about 2½ feet thick, of approved material, in cofferdams, worked along the sides above the protective deck the entire length of the vessel. The cellulose is to be supplied by the contractor, and shall be fireproofed by a process satisfactory to the Department. The cofferdam will be packed by the contractor.

No woodwork is to be incorporated in the hull or fittings, except such as is indispensable, and all woodwork used in the construction of the hull and fittings, except the battens for electric-light wires, shall be fireproofed by a process satisfactory to the Department.

Only the superstructure deck and the main deck outside of the superstructure are to be of wood, which are to be laid on complete metal decks.

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Special care is to be observed to provide for commodious and well-ventilated engine and fire rooms, and the system of ventilation is to extend to all living spaces, coal bunkers, ammunition rooms, holds, storerooms and gun casemates.

* * * * *

Arrangements shall be made for cooling the magazines, or certain specified ones, by means of compressed air, by connec-

tion with the refrigerating plant, or by other approved means, and, if required, the magazines, or certain specified ones, shall be further insulated by an approved non-conductor of heat.

* * * * *

The coal bunkers must be well arranged, with ample facilities for rapid coaling and for supplying the fire rooms with coal, and small sub-divisions and inaccessible spaces are to be avoided as far as practicable.

Approved means shall be provided for opening and closing by power, as well as by hand, such watertight door as may be designated, with suitable devices for operating at each door and from a central station, and for indicating at such station whether the doors are open or closed.

Each vessel is to be fitted for a flagship, and ample provision must be made for the accommodation of the full complement of officers and crew, which will comprise—

1 flag officer,	8 warrant officers,
1 commanding officer,	525 crew, including marines.
1 chief of staff,	———
18 wardroom officers,	564 total.
10 junior officers,	

A small and compact laundry, with capacity to wash for about 75 men, is to be provided and equipped with the following laundry machinery of approved type, viz: One washer, one hydro-extractor, and one mangle; also with two soapstone set tubs, one soap tank, ironing table, and drying room. The line and counter shafting, for the above-named machinery (the former to be run by electric motor) is to be supplied by the contractor, who will also provide all necessary steam and water connections.

* * * * *

There shall be suitable separate cabins for the flag officer and for the commanding officer. Mess rooms for the wardroom, junior and warrant officers, and the following staterooms will be required, in addition to those in the cabins, viz:

For wardroom officers, 18.

For junior officers, 5.

For warrant officers, 8.

Also one, with toilet room, for chief of staff.

The following machine tools of approved make and design will be furnished and installed by the contractor in the general workshop, with proper line and counter shafting (the former to be run by electric motor), and all necessary and usual spare parts and tools, viz:

One screw-cutting back-geared gap lathe, to swing 30 inches over the ways and to take 10 feet between centers.

One 14-inch screw-cutting back-geared lathe to take 4 feet between centers.

One column-shaping machine, 26-inch stroke, 26-inch traverse.

One double-geared drilling machine, to drill up to 1½ inches, 18 inches from edge of work.

One 16-inch sensitive drill.

One No. 1 Universal milling machine.

One combined hand punch and shears, with 6-inch shear blades, to cut ¾-inch round iron and ⅝-inch steel plate, and to punch ⅝-inch holes in ⅝-inch plate.

One emery grinder, two 12 by 2 inch wheels.

One 30-inch grindstone.

Six bench vises.

A suitable place will be provided for a blacksmith shop, suitable tools for which will be furnished and installed by the contractor.

Ample facilities for convenient and rapid coaling must be provided by means of steam winches, booms, and derricks; also a sufficient number of well-placed chutes and coaling ports.

The following auxiliary machinery of approved make and design, in addition to that pertaining to the main engines and their dependencies, to be operated by steam power, is to be supplied and installed by the contractor, complete with all the necessary steam and exhaust piping and fittings, viz:

Steam steering engine.

Anchor engine and capstan.

Ash-hoisting engines in each fire room.

Dense-air ice plant of approved type, with a cooling effect of 2 tons of ice per day.

Evaporating plant, to consist of four units, each having a ca-

pacity of 5,250 gallons of fresh water a day; also a distilling apparatus capable of condensing at least 10,000 gallons of potable water a day.

Five deck winches of 30 horsepower each; each winch to have two hoisting drums, either of which may be operated independently of the other.

Engines of 50 horsepower for each of the heavy boat cranes.

The following auxiliary machinery is to be supplied and installed by the contractor, and is to be operated by electric power, viz:

Ammunition hoists.

Blowers for hull ventilation.

The line shafting for the machinery in the general workshop will be electrically driven, the contractor supplying the motors therefor.

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The engines will be of the vertical, twin-screw, four-cylinder, triple-expansion type, of a combined I.H.P. of 21,000. The steam pressure will be 250 pounds. The stroke will be 45 inches. The ratio of high pressure to low-pressure cylinder will be at least 1 to 7.30, and the diameters will be sufficient for the required I.H.P. at about 133 revolutions per minute. Each engine will be located in a separate watertight compartment. They will be provided with all the necessary auxiliaries and accessories in accordance with the latest practice of the Bureau of Steam Engineering.

There will be sixteen boilers of the straight water-tube type, placed in four water-tight compartments. They will have at least 1,400 square feet of grate and 58,800 square feet of heating surface, and must be able to furnish steam for the main engines and all the necessary auxiliary machinery with an average air pressure in the ash pits of not more than one inch of water. All the necessary auxiliaries and accessories will be provided for the efficient working of the boilers.

There will be four funnels, each 100 feet high above the base line.

Fifty tons of fresh water will be carried on trial in the double-bottom or reserve tanks for use of the water-tube boilers.

The vessel is to be heated with steam throughout, and the usual steam-heating system will be provided and installed by the contractor.

The weight of all machinery and tools, stores and spare parts will be about 1,900 tons. This weight must include all articles, irrespective of name or use, coming under the cognizance of the Bureau of Steam Engineering, including water in boilers, condensers, piping, etc., but excluding the reserve feed water in the double bottom or tanks.

The armanent will be as follows:

Main battery: fourteen 6-inch breech-loading rapid-fire rifles of 50 calibers in length.

Secondary battery: Eighteen 3-inch breech-loading rifles (14-pounders) of 50 calibers in length; twelve 3-pounder semi-automatic guns; four 1-pounder automatic guns; eight 1-pounder rapid-fire guns; two 3-inch field guns; two machine guns, .30 caliber; eight automatic guns, .30 caliber.

The electric generating plant will consist of five units, each unit to consist of an engine and dynamo mounted on a combination bedplate. Two units shall have a rated output of 1,250 ampères each at 80 volts, and three units shall have a rated output of 625 ampères each at 80 volts. The total weight of the five units complete must not exceed 100,000 pounds. The total weight of the whole electric installaton, including dynamos, engines, bedplates, all fittings, wiring, tools, stores, instruments, and six search lights, must not exceed 123 tons.

* * * * *

Summary of principal weights, Department's design:

	Tons.
Guns, mounts, shields, etc	270.9
Ammunition, ordnance stores, and outfit	372.2
Steam engineering complete, with water in boilers, condensers, piping, and stores, etc., not to exceed	1,900.0
Reserve fresh water for steaming purposes (full supply)	150.0
Total coal	1,500.0
Boats and outfits	31.5
Masts and spars	29.7

	Tons.
Equipment, complete (including anchors, chains, electric plant, etc.)	252.7
Miscellaneous stores and water (full supply)	58.3
Officers, crew and effects	73.6
Provisions, clothing and small stores (full supply)	145.7
Total protection, including armor, bolts, nuts, splinter bulkheads and cellulose	854.0

Wyoming.—The *Wyoming* was launched at the Union Iron Works, San Francisco, on September 8th, in the presence of His Excellency the Governor of Wyoming and an enormous crowd of spectators. The *Wyoming* is considerably less in dimensions than the *Monterey*, though both are of the same type. The *Monterey* is 255 feet in length, and is 59 feet beam, with a depth of 14 feet 10 inches. The *Wyoming's* dimensions are: Length, 252 feet; beam, 50 feet, and mean draught, 12 feet 6 inches. The side armor of the former vessel is 13 inches in greatest thickness, tapering to 8 inches at the ends; of the latter, 11 inches, tapering to 3 and 5 inches. Displacement: *Monterey*, 4,084; *Wyoming*, 3,235; horsepower engines, 5,244 and 2,400 respectively; speed, 13.6 and 12.

The *Wyoming* will have a speed of about 12 knots with about 2,400 H.P., using steam at 250 pounds pressure. The engines are vertical, triple-expansion (twin screw) 17 by 26½ by 40 inches, and 24-inch stroke. The boilers (water-tubular) have 216 square feet grate surface, and 9,400 square feet heating surface.

The armament of the *Monterey* consists of two 12-inch and two 10-inch main battery, six 6-pounders, four 1-pounders, two Gatlings. Of the *Wyoming*, two 12-inch, four 4-inch rapid fire, two 6-pound semi-automatic, four 1-pound automatic, and additional four automatic 1-pounders have been authorized.

Arkansas. — The harbor-defense monitor *Arkansas* was launched at Newport News, on Saturday, November 10. Miss Bobbie Newton Jones, daughter of Gov. Jones of Arkansas, christened the vessel. There is a single-balanced turret on the center line forward, with an inclined top, and made of steel 9 inches thick. This turret will contain two of the new high-power 12-inch breech-loading rifles. Besides this armament, the *Arkansas* will carry four 4-inch rapid-fire guns, three 6-pounders, and four

1-pounders. The *Arkansas*, when complete, will draw 12 feet 6 inches of water, on a displacement of 3,235 tons. Eleven inches of steel armor protect the sides of the monitor above the water line, and there will be a protective deck for the full length of the vessel 1½ inches thick. Twin-screw engines will drive the monitor, at a maximum speed of 11½ knots, amply sufficient for a harbor-defense vessel, and electricity will be employed for turrent-turning gear, ammunition hoist, ventilating apparatus, and all the secondary machinery. Unlike the old type of monitors, the *Arkansas* will have staterooms above deck for the officers and crew.

Nevada.—The monitor *Nevada* was successfully launched from the yards of the Bath Iron Works, Bath, Me., on November 24, in the presence of a large crowd of spectators. Miss Annie Curtiss Boutelle, daughter of Congressman C. A. Boutelle, christened the vessel.

The official dimensions of the *Nevada* are: Length, 252 feet; extreme beam, 50 feet; draught, 12 feet 6 inches; displacement, 3,234 tons.

The batteries consist of two 12-inch guns, four 4-inch rapid-fire guns, three 6-pounders and four 1-pounders.

The turrets for the 12-inch rifles are of the balanced sort. Electricity will be utilized throughout in handling turret-turning gear, ammunition hoists and ventilating blowers.

The contract speed for this monitor is 11.5 knots. The plan of hull of the craft resembles that of the *Monterey*.

There will be two sets of engines of the vertical, inverted-cylinder, direct acting, triple-expansion type, placed in one compartment. They are designed to develop a collective horsepower of 2,400 and in attaining this development the engine must be turned up to 200 revolutions per minute. Steam will be taken from water-tube boilers and will be carried for a full working power at 250 pounds pressure to the square inch. The four water-tube boilers provided for each ship will be placed in one fore-and-aft compartment, two boilers on each side, and will furnish a total heating surface of 8,800 square feet and at least 200 square feet of grate surface.

The armor protection of the new craft will consist of an 11-inch belt on the hull, extending the full length of the vessel, and tapering at the top to 5 inches and at the bottom to three. The turret armor will be 10 inches thick, while the conning tower will have 7 inches of armor and the protective deck will be 1½ inches thick.

Lawrence.—The U. S. torpedo-boat destroyer *Lawrence* was successfully launched at Weymouth, Mass., November 7, being christened by Miss Ruth Lawrence, of New York, a descendant of Capt. James Lawrence, U. S. N., who commanded the *Chesapeake* in her action against the British vessel *Shannon* off Boston on June 1, 1813.

The general dimensions of the *Lawrence* are: Length over all, 246 feet 3 inches; extreme beam, 22 feet 3 inches; depth of hull, 16 feet 8 inches; greatest draught, 8 feet 6 inches. She will have a displacement when loaded of 470 tons, and an indicated horse-power of 8,400, at about 360 revolutions per minute of the engines. The hull is of steel construction, all parts of which, to a height of one foot above the water line, are galvanized. She was constructed by the Fore River Engine Company. The engines are twin-screw triple-expansion 22 inches by 31 inches by (2) 34 inches by 20-inch stroke. The boilers are of the Express type, water-tubular, with 320 square feet of grate surface and 18,000 square feet of heating surface.

Perry.—The torpedo-boat destroyer *Perry* was launched at the Union Iron Works, of San Francisco, Cal., on October 27. This is one of the sixteen destroyers ordered by the Government two years ago, and is the first to be launched on the Pacific coast. Her general dimensions are like those of the rest authorized, or water-line length, 245 feet; extreme beam, 23 feet; draught over top of her twin screws, 8 feet; speed, 30 knots; coal carried, 100 tons; limit of cost, $295,000, with no bonus for excess speed. She will carry two 3-inch rapid-fire guns, five 6-pounder guns, and two 18-inch torpedo-tubes.

Submarine Torpedo Boats.—Contract has been made for the construction of six Holland submarine boats, at a cost of

$170,000 each. The contract has been sublet as follows: To Union Iron Works, the *Grampus* and *Pike;* to Lewis Nixon, the *Adder, Moccasin, Porpoise* and *Shark.*

A further contract has been made with the Holland Co. to build one more boat to replace the *Plunger.*

Dahlgren and Craven in Collision.—The torpedo-boat destroyers *Dahlgren* and *Craven* collided off Castle Hill, R. I., October 16, during a gale, and had a narrow escape from foundering while en route to Portsmouth, N. H. The vessels left Newport in company, and when near Fort Adams the *Dahlgren's* engines were stopped for some reason, the *Craven* passing her. The *Dahlgren,* after getting under way, attempted to cross the bow of the *Craven,* and in doing so the *Craven* struck the *Dahlgren* 25 feet abaft the stem, the impact twisting the *Craven's* bow until it pointed toward the stern. As the *Craven* was making water rapidly, her crew prepared to leave the craft, believing it was sinking. In attempting to lower the boat one of her crew fell overboard, and he was in the water nearly half an hour and was nearly exhausted before being picked up by one of the boats from the *Dahlgren.* The *Craven,* with one engine going ahead and the other reversed, in order to overcome the twist in the bow of the boat, succeeded in reaching her slip at the torpedo station. The *Dahlgren,* only slightly damaged, kept close to the *Craven* and attempted to make the slip next to her. A northerly gale was blowing at the time, and the *Dahlgren* plunged into the stern of the *Craven,* ripping away everything for ten feet. The *Dahlgren* made a second attempt to reach the slip, but again was blown out of her course, and a second time rammed the *Craven.* The *Dahlgren* made no further attempt to reach the slip, but cast her anchors with four fathoms of cable. They failed to hold, and the *Dahlgren* drifted through the harbor, running onto the mud flats at the south end. Lieut. William G. Miller was in charge of the *Dahlgren* and Lieut. Ford H. Brown of the *Craven.*—"Army and Navy Journal."

The Loss of the Converted Cruiser Yosemite.—Advices have been received from Guam that the island was visited by a terrific typhoon on November 13, which wrecked thousands of

houses, among them being the headquarters of Governor Schroeder.

The towns of Indrajan and Terra Foro were swept away, and it is estimated that hundreds of the native population in various parts of the island met their death. The cocoanut crops were rendered absolutely worthless, and the vegetation of the island killed by salt water. The storm came up in the forenoon and swept across the island with amazing rapidity.

The United States auxiliary cruiser *Yosemite*, which was anchored adjacent to the collier *Justin*, dragged her anchor and was driven aground a hundred and fifty yards from the reef. Her bows were crushed in. A launch with a crew of five men had previously left the ship to endeavor to find a safe anchorage for the vessels, the indications being that the anchors would not hold where she was. The men were not seen after they left the ship, and it is practically certain that the heavy sea which the storm kicked up swamped the launch. The bodies of Coxswain F. Swanson and Seaman George Anhel were recoverd after the typhoon subsided. The storm veered around after the *Yosemite* grounded, and she was driven off and carried on to the Somaye cliffs, where her rudder and propeller were broken.

Boatswain Sweeney and twelve of the crew volunteered to take a life line ashore. A boat was lowered for this purpose, but it was immediately swamped by a big sea that swept over it. All its occupants were carried away from the boat, but they miraculously succeeded in reaching the land after an hour's struggle with the waves.

Meanwhile the atmosphere had become of inky blackness, and the *Yosemite*, which had again blown off the land, was drifting helplessly before the gale, which was blowing at the rate of 100 miles an hour. All hope of saving the cruiser was abandoned, and the officers and crew prepared for the death they thought was surely staring them in the face. A majority of the boats had been either smashed or carried out of the davits, and those that were left could not accommodate half the crew, even had it been possible for them to live in the enormous seas that were by this time running. A sea-anchor had been put out, and this held

the *Yosemite* up into the teeth of the storm and prevented her from falling off into the trough of the sea.

She rode to this anchor and drifted rapidly until 6 o'clock in the evening, when the weather cleared. The cruiser was then sixty miles north and forty miles west of Guam. She had labored so heavily that she had sprung a leak, and all hands, including her commander, Lieutenant Breif, and the chief engineer, worked heroically in trying to clear her of the water that was pouring into and rapidly filling the forward compartments. After the violence of the storm had subsided efforts were made to start the engines. They were finally got to work, and the *Yosemite*, with her damaged propeller, struggled landward at the rate of two knots an hour. The water kept gaining in the hold and the ship was gradually sinking.

At 1.30 P. M., November 15, the *Justin*, which had started in search of the *Yosemite*, picked her up and attempted to tow her back to Guam. Two hawsers were broken, and it was then decided that it was impossible to take her into port. The cruiser was then scuttled, after which she was abandoned, all hands going aboard the *Justin*. The *Yosemite* sank bow first at 3 o'clock and the *Justin* stood away for Guam.

The *Yosemite* was first commissioned April 13, 1898. She had been a station ship at the Island of Guam since the days of the Spanish war. She is a sixteen-knot converted cruiser of 6,179 tons displacement, and is provided with a main battery of ten five-inch rapid-firing guns and a secondary battery of six six-pounders and two Colt rifles. She has a complement of 18 officers and 267 men. Her measurements are 389 feet 2 inches in length by 48 feet extreme breadth. She has a mean draught of 20 feet 1 inch.

Just before the outbreak of the Spanish-American war the vessel was purchased by the Navy Department, fitted as an auxiliary cruiser, and rendered effective service as one of the patrol fleet, being manned in part by the Michigan naval militia.

According to the naval register, the *Yosemite* was manned by 18 officers and 267 men.

ENGLAND.

New Ships.—The official list of the ships ordered to be commenced at once has been just published, and includes two battleships, six armored cruisers, two second-class cruisers, and a couple of sloops:

Battleships: *Queen*, to be built at Devonport; *Prince of Wales*, to be built at Chatham.

Armored cruisers: *Cornwall*, to be built at Pembroke; *Suffolk*, to be built at Portsmouth. The four others have been given to private firms as follows: *Berwick*, to Beardmore & Co., Ltd.; *Cumberland*, to the London and Glasgow Shipbuilding Co.; *Donegal*, to the Fairfield Shipbuilding Co., Ltd.; *Lancaster*, to Armstrong, Mitchell & Co., Ltd.

Second-class cruisers: *Challenger*, will be built at Chatham; *Encounter*, will be built at Devonport.

Sloops: *Odin* and *Merlin*, will be built at Sheerness.

The details of these ships have not yet been published, with the exception of the following general data of the *Lancaster*, which will be applicable to all of the armored cruisers:

Length over all, feet	463½
between perpendiculars, feet	440
Breadth, extreme, feet	66
Draught, feet	24½
Displacement, tons	9,800
Speed, knots	23
Indicated horsepower	22,000
Length of trial (normal draught), hours	8
Engine Cylinders:	
H.P., diameter, inches	37
I.P., diameter, inches	60
L.P. (2), diameter each, inches	69
Stroke, inches	42
Number of boilers	31
Type of boilers	Belleville.
Coal capacity, normal, tons	800
full, tons	1,600

Armament: Fourteen 6-inch Q. F., four in pairs in barbettes, and ten in casemates; ten 12-pounders; three 3-pounders; 2 torpedo tubes (submerged); 8 machine guns.

Essex Class of Cruisers.—At present there is considerable interest in the construction of cruisers—four nations having under construction or in contemplation a type of vessel which may be regarded as of the same class. These are the *Essex* class of the British navy, the *Bayan* class of the Russian navy, of which the *Variag* is one; the *Prince Heinrich* class of the German navy, and the three protected cruisers provided for in the last naval bill of Congress. The four armored cruisers of the *Essex* class—*Essex*, *Kent*, *Bedford* and *Monmouth*—although a great deal heavier than the *Bayan* and *Prince Heinrich*, are essentially of the same type. The principal particulars of the type are as follows:

Displacement, tons	9,800
Length, feet	440
Beam, feet	66
Draught, mean, feet	24½
Armament—Fourteen 6-inch 45-caliber; ten 12-pounders; three 3-pounders, and an indefinite number of Maxims, probably eight.	
Torpedo-tubes	Two 18 inch submerged.

The *Essex* has the high forecastle common to all cruisers of the British navy nowadays, and her 12-pounder battery is amidships. The high forecastle is absolutely indispensable to a swift cruiser—otherwise they take in too much sea forward. The 6-inch guns are thus disposed: Two in a turret forward, two in a turret aft, six on the main deck amidships and four on the upper deck above the forward and after main-deck guns. All these pieces are in armored casements. Six 12-pounders are on the upper deck amidships, two under the forecastle forward, two on the main deck aft. This is the armament of the *Essex* in the original design, though there are rumors about installing a 7½-inch quick-firer. All fourteen 6-inch guns are protected.

The armor of the *Essex* is thin and extensive. There is a belt about 250 feet long amidships of 4-inch Krupp cemented armor, and this belt extends right up to the main deck. It is continued to the bow at a thickness of 2 inches. Aft it is terminated by a 5-inch bulkhead. A curved protective deck runs throughout the length of the ship and reinforces the water-line protection. The

space above and below is filled, as usual, with coal bunkers. The casements are 4 inches thick; the turrets, 5 inches; but for these nickel armor, not Krupp, is employed. It has less resisting power, but is more easily worked. These turrets have short-armored bases of 5 inches thickness, with armored hoists going down the main deck. These turrets are distinctive and, so far, peculiar to the *Essex* class. To begin with, the guns in each can be trained together and fired as one piece or else used independently. These guns and their mountings are to be furnished by Vickers, Sons & Maxim, Ltd., who have brought out the design. There is a single hoist to a pair of guns worked by an electric motor. It serves with very great rapidity, and when in full working order, delivers every alternate charge to the port gun, which has a special small motor to catch it as it comes up the endless chain. This arrangement can be easily disconnected—the hoist is then made to serve one gun only.

The *Essex* will have the usual rig for modern British cruisers, a search-light platform on each top, but no fighting tops for guns. There will be three funnels. The engines are designed to give 23 knots speed with 22,000 horsepower. The boilers will be Belleville of the latest pattern fitted with economizers. Comparing the *Essex* with the *Bayan* and *Prince Heinrich* as regards powers of offense and defense the following is found:

	Essex.	*Bayan.*	*Prince Heinrich.*
Displacement	9,800	7,800	8,868
Horsepower	22,000	17,000	15,000
Speed (maximum), knots	23	21	20
Main armament	Nil.	Two 8-inch.	Two 9.4-inch.
Secondary armament	Fourteen 6-inch	Eight 6-inch.	Ten 6-inch.
Tertiary armament (protected)	Nil.	Eight 3-inch.	Nil.
Tertiary armament (unprotected)	Ten 3-inch	Twelve 3-inch.	Nil.
Small pieces	Three 3-pdrs. and Maxims.	Seven 3-pounders.	Ten 1-pdrs., four Maxims.
Submerged tubes	Two.	Two.	Three.
Above-water tubes	Nil.	Three (2 protected)	One.
Armor belt (amidship)	4-inch.	8-inch.	4-inch.
Armor belt (bow)	2-inch.	4-inch.	3-inch.
Lower-deck armor	4-inch.	3-inch.	4-inch.
Protection to main armament		7-in. on each turret; 3-in. hoists to these.	6-inch.
Protection to secondary armament	Four 6-in. in turrets with 5-in. armor. Ten 6-in. in casem'ts with 4-in. armor.	3-in. on 3 redoubts.	6-in. on turrets and redoubt.
Protection to tertiary armament	Nil.	3-in. on eight of them	Nil.

The *Essex* may, therefore, be regarded as first in speed and defense, with the *Prince Heinrich* first in offense. The *Bayan* is second in speed, though whether this advantage will be maintained in actual service is questionable, as the Russians are not strong in their engineering department. The American contribution to the *Bayan* type, the *Variag*, is, however, the fastest cruiser afloat.—" Marine Review."

Spartiate.—The first-class protected cruiser of 11,100 tons displacement and 18,000 I.H.P. has begun a series of trials, the first of which terminated November 9th. On this but ten of the Belleville boilers were used, developing 3,834 I.H.P. and securing 12.3 knots speed with 70 revolutions of the engines per minute. The coal consumption was 1.83 pounds per I.H.P per hour. The next trial is to be 30 hours at four-fifths power, and finally an eight-hour full-power forced-draft test.

Viper.—The successes which the *Viper* and *Cobra* have attained in being the fastest vessels afloat has led to criticisms being passed as to their coal consumption and radius of action. This has led the Hon. C. A. Parsons to reply, and, describing the trials through which the *Viper* has already passed, he says:

" They have included a coal-consumption trial at the contract speed of 31 knots, the mean speed during the three hours being 31.118 knots, and the coal consumption, as determined from the usual hull-resistance experiments, 2.38 pounds per I.H.P. per hour, the contract being not to exceed 2.5 pounds per I.H.P. per hour. The power required for this speed is about two-thirds of the maximum for which the engines and boilers were designed, the maximum being upwards of 12,300 I. H. P., which power has been realized on preliminary, but official, full-power contractors' trials, the maximum mean speed then reached being 36.858 knots.

" The second official trial was a three hours' coal-consumption trial with the air pressure limited to 3½ inches, and carrying the full weights as determined by the previous trial under the usual Admiralty conditions. A mean speed of 33.838 knots was maintained with about four-fifths of the maximum power, the coal consumption being at the rate of 2.49 pounds per I.H.P. per hour.

"The third official trial was the usual twelve hours' coal-consumption trial at cruising speed. In recent years the stipulated speed on this trial has been increased from 13 knots to 15 knots (this is mentioned as the coal consumption of the *Viper* at 13 knots is only about two-thirds of that at 15 knots.) The ascertained coal consumption of the *Viper* on this trial was at the mean rate of 27 hundredweight per hour. It should be mentioned, however, that the *Cobra*, the second fastest ship afloat, with the same sized turbine engines and similar in every respect to the *Viper*, but loaded to service conditions and having a displacement of 442 tons, or about 60 tons greater than the *Viper*, consumed on her official 15-knot trial only 24.58 hundredweight per hour. In her case the port engines only were used, the starboard engines being dragged round by the propellers, and it is probable that the *Viper*, with her less displacement, would, when using one set of engines only, burn less coal than the *Cobra*. As, however, it would be instructive to ascertain this result definitively, it is probable that this test will be made at an early date."

As regards coal consumption, of which comparisons have been instituted between the *Viper* and a torpedo-boat destroyer, *Albatross*, of about the same displacement, Mr. Parsons points out:

"In the case of the *Viper* all of the underwater fittings, such as rudder, the shafts and brackets, for carrying the propeller shafts, the propeller blades and bosses, and so forth, have all been designed to stand the highest speed of over 36 knots, and are of considerably larger and heavier scantlings than usual, and in consequence offer increased resistance to their passage through the water. So much for the shipbuilder's point of view. . . . If the *Viper's* propellers (adds Mr. Parsons) had been designed for a maximum speed of only 32 knots (the contract speed of the *Albatross*) they would undoubtedly have given superior results at about 31 knots."

In concluding his reply, Mr. Parsons adds:

"In the specification of most Admiralties for torpedo boats and destroyers, the question of speed is placed in the fore part and enforced under heavy penalties, culminating with the option of complete rejection of the vessel should the deficit reach

more than two or three knots below that contracted for. The coal consumption is relatively placed in the background. This being so, the designers of the *Viper* may be pardoned if they have made every effort to attain the highest possible speed and have not given sufficient consideration to the question of obtaining the utmost economy of fuel of which turbines are capable; but in this, their first, destroyer they can at least claim to have beaten all records of speed by a long interval, and in their future ships they will be able to show by how much they can beat all records as to economy in coal."

Messrs. Thornycroft and Co., in commenting on the communication of Mr. Parsons, from which we have quoted, says:

"When a comparison is made between the actual distance which one ton of coal will carry the *Albatross* fitted with reciprocating machinery and the distance which the same quantity of coal will carry the *Viper*, which is of the same displacement, but is fitted with turbine machinery of much greater maximum power, it is found that, while one ton takes the *Albatross* 22.6 nautical miles, at a speed of 15 knots, it will only carry the *Viper* rather less than 12. Moreover, the space demanded by the machinery in the more highly-powered vessel leaves less room for bunkers, so that the radius of action of the *Albatross* at cruising speed is probably more than double that of the *Viper*."

FRANCE.

Chateaurenault.—This new first-class cruiser has been continuing her trials off Toulon successfully. With the engines developing 14,000 I.H.P., the engines made 112 revolutions, giving a speed of 20.5 knots with a coal consumption of only 660 grammes per H.P. per hour; with the engines developing 18,000 I.H.P., and making 120 revolutions, she averaged for two hours 22.6 knots; it is, therefore, considered certain that with the engines developing the full 23,000 I.H.P. her contract speed of 23 knots will be exceeded.

Hoche.—This first-class battleship has undergone fresh trials of speed at Brest, her first trials having proved unsatisfactory owing to insufficiency in the steam supply. In the first trials the

engines developed 10,912 I.H.P., against 12,000 expected, and she attained a speed of 15.92 knots, with a consumption of coal which would have emptied her bunkers in a run of 803 miles at that rate of speed. In the new trials her engines developed sufficient power to give her a speed of 16 knots, the consumption of coal being smaller, so that her supply would last her for a distance of 1,292 miles at the 16-knot rate of speed. This is considered satisfactory. The chief effect of the transformation has been to decrease her draught. She used to be called a submarine battleship, but now she has more than 2 feet of freeboard. This has been effected by reducing her superstructure, her after military mast, and the total weight of her artillery. It is expected that she will join the Northern Squadron in October.

The Loss of the Framée.—Vice-Admiral Fournier, commanding the Mediterranean Fleet, has forwarded his report on the unfortunate sinking of the *Framée* (destroyer) by collision with the flag-ship *Brennus*, off Cape Trafalgar, on the night of 11th August, while the fleet was steering for the Straits of Gibraltar, being at the time on passage from Royan to Toulon.

The night was clear and calm, with a bright moon, the fleet proceeding in line ahead, the station of the *Framée* being off the starboard beam of the flagship, in line with the after bridge. Suddenly, while a signal was being made to her shortly before midnight, the *Framée* was observed to be steering at full speed for the *Brennus*. Everything possible was done on board the latter ship to avert a collision by the officer of the watch, but without success, the *Framée* striking the *Brennus* on the starboard bow and sinking almost immediately.

The fleet stopped, and boats were lowered, but, notwithstanding the clearness of the night and smooth sea, only 14 of the crew of the *Framée* were saved. Her officers were all on deck at the time, but not one was rescued. A quartermaster of the *Brennus*, while the vessels were still in contact, held out his leather belt to Commander De Mauduit Du Plessix, of the *Framée* but that officer turned away, saying simply '*Tout à l'heure*,' and almost immediately afterwards the *Framée* sank, and he went down with her. As no officers were saved, it is difficult to ac-

count satisfactorily for the disaster, but it is stated by one of the survivors that he heard the order "20° to port!" given by Commander Du Plessix, and it is supposed that by some mistake the helm was put the wrong way, and that the mistake may be due to the system of communication for the transmission of orders, which is said to be defective.

Launch. — The torpilleur-de-haute-mer *Audacieux* was launched on the 29th August at Nantes. She is of the same name and class as the torpedo boat which sank off the coast of Corsica in 1896 in consequence of a collision with the torpedo boat *Chevalier*. She is of the *Cyclone* type. Her dimensions are as follows: Length, 147 feet 8 inches; beam, 15 feet; draught, 8¼ feet; displacement, 150 tons; her engines are to develop 4,250 I.H.P., giving a speed of 30 knots; armament, two 1.85-inch guns and two torpedo tubes; radius of action, with 18 tons of coal, 1,800 miles at 10 knots, or 200 miles at 30 knots.—"Le Yacht and Le Temps."

Wreck of the Bouet Willaumez.—The torpilleur-de-haute-mer *Bouet Willaumez* belonging to the Défense-Mobile of Cherbourg, while cruising off that port in foggy weather, struck on a rock near Bréhat and sank, the crew being saved by torpedo-boat No. 108, which was in company with her. She was used with other torpedo boats for training a special corps of petty officers, called torpedo-boat pilots, so that every torpedo boat in time of war may have a thoroughly competent pilot on board, fully acquainted with the numerous reefs and banks which surround the French coast. All attempts to refloat the boat have been given up as hopeless.—"Journal Royal United Service Institution."

GERMANY.

The Prinz Heinrich.—This triple-screw armored cruiser is completing for sea and is apparently intended as a standard type for German armored cruisers. Unlike the *Fürst Bismarck*, she is not a merely weak battleship, but a cruiser pure and simple, built chiefly with an eye to cruiser work. Her principal dimensions, etc., are as follows: Displacement, 8,868 tons; length, 394 feet; beam, 65½ feet; draught (mean), 23¾ feet; armament,

two 9.4-inch (24 centimeter); ten 6-inch (15 centimeter); ten 1-pounders; four Maxims. Torpedo tubes: one bow, submerged; two broadside, submerged; one, stern, above water.

The armor is distributed as follows: There is a complete water-line belt of Krupp armor 4 inches to 3 inches in thickness, reinforced by a protective deck 2¾ inches thick on the slopes. Above the belt there is a redoubt about 165 feet long, and of about the same thickness as the belt. Above this again is the main deck redoubt, 6 inches thick, in which six of the 6-inch guns are mounted. Both redoubts have 6-inch bulkheads, while screens will be placed between the guns in the upper one. On top of the upper redoubts there will be four turrets, oval in shape, each 6 inches thick, and carrying a 6-inch gun. Forward and aft are barbettes, each mounting a single 9.4-inch gun, and protected by stout shields. These barbettes will have a total height of about 9 feet, will be 6 inches thick, with armored hoists descending to the protective deck. The conning tower, which will have a thickness of 6 inches or more, will have an armored tube of communication.

It will be noted that the *Prinz Heinrich* has little in common with the *Fürst Bismarck*. The later has no protection to the lower deck, and has her armament a good deal more distributed. The *Prinz Heinrich*, on the other hand, is a direct adaptation of the French *Brennus*, and closely resembles that ship both in the disposition of armament and in the system of armoring. The high forcastle and the bow submerged tube are the only distinctly German features in the design, and there is nothing of the *Fürst Bismarck* about her save in general outline.

Like all German ships, the *Prinz Heinrich* will have no wood used in her construction, unless the latest discovery, or supposed discovery, that woodless ships are very unsanitary, causes some departure from a well-established German custom. The estimated indicated horsepower is 15,000, which is expected to give a speed of 20 knots. There are three screws, and the deadwood aft is very much cut away, after the *Elswick* model, in order to make the ship relatively handy. This, for a cruiser, she should

be, as she has a very fair proportion of beam to length. The boilers selected for her are of the *Dürr* type.

It may be of some interest to compare the *Prinz Heinrich* of 1900 with the *Edgar* class of 1890 and the *Orlando* of 1886, since the armaments in each case are practically identical, though the German 9.4-inch gun is, of course, a far more powerful gun than the old 9.2-inch in the other ships. This, however, is more an accident of date than anything else, and, in so far as the modern guns are all heavier, enforces rather than detracts from the point we wish to make—the difference between what was demanded then and now for a ship carrying a more or less standard armament for a first-class cruiser in the way of protection and speed.

——	*Orlando.*	*Edgar.*	*Blake.*	*P. Heinrich.*
Date of launch	1886	1890	1891	1890
Displacement, tons	5,700	7,700	9,000	8,868
Length, feet	300	360	375	394
Beam, feet	56	60	65	65½
Draught, feet	26	26	27½	24¾ (mean)
Armor belt, inches	10	Nil.	Nil.	4
lower deck, inches	Nil.	Nil.	Nil.	4
battery, Q. F., inches	Nil.	6 in. on 4 guns.	6 in. on 4 guns.	6
big guns, inches	4½ in. shields	4½ in. shields.	4½ in. shields	6
deck, inches	3	5	5	2¾
I.H.P. *circa*	9,000	10,000 to 12,000	20,000	15,000
Speed, trial	18.5 to 19	19.5 to 20	21.5	20 (est.)
Actual sea-speeds at a push	15.5 or less	19.5	20.5	.....

The *Orlando*, the only cruiser carrying belt armor except the *Prinz Heinrich*, has less than this ship, and is altogether smaller. She belongs to the era of "paper" ships, and to this day, with her 10-inch "armor belt"—all under the water-line, by the way—and heavy armament, is a grand ship for the makers of statistics. The *Prinz Heinrich* could probably tackle three of her. The *Edgar* and *Blake* are more interesting comparisons. All the class are as good ships as ever were built for speed. Although now nearly ten years old, they are all excellent steamers, and in the recent maneuvers kept station at 19 knots without the least

difficulty. Ten years have seen far less advance in speed than people are prone to imagine. It is exceedingly doubtful whether there is a warship in the world that would beat the *Edgar* class in a thirty hours' race. Probably the Russian cruiser *Variag*, recently built at Cramp's, Philadelphia, might. The advance of recent years has been in protection, and the problem to-day is to pile on armor without deteriorating the other cruiser essentials.

ITALY.

Three New Battleships, of 12,000 tons displacement and 19,000 I.H.P., having a designed speed of 21 knots, will be begun early in 1901 at Spezia, Castellamare and Venice.

Varese.—The armored cruiser *Varese* made her first natural-draft trial on October 20, when for six hours the average revolutions were 96 per minute, giving 9,200 I.H.P. and a speed of 18.3 knots. The coal consumption was 2.07 pounds per I.H.P. per hour, which was considered very satisfactory. The *Varese* is a 7,400-ton cruiser with full power of 13,500 I.H.P. and 20 knot speed.

JAPAN.

Mikasa.—Perhaps the most interesting feature in the design of the *Mikasa*, the Japanese battleship launched by Messrs. Vickers, Sons and Maxim, Limited, at Barrow-in-Furness, on Thursday, November 8, 1900, is in the distribution of her armor, several modifications in recent practice having been made in the design. In giving a brief description we, therefore, give first consideration to this. As in the two earlier Japanese battleships built in this country, the *Asahi* and *Hatsuse*, the water-line armor belt extends from end to end, instead of terminating at the ends of the citadel, and thereby leaving about 80 feet from the stern unprotected. The main belt has a width of 7 feet 9 inches, of which 5 feet 3 inches is below the water-line. The main belt armor in the way of the citadel, for 156 feet in length, is 9 inches in thickness, beyond this at either end—to some distance past the forward-and-aft 12-inch armored transverse bulkheads—it is 7 inches in thickness, assisted by additional 1½-inch nickel-steel plates placed on the slope of the protective deck, so

as to make the resistance equivalent to 9-inch Krupp-Harveyed armor, for the whole length of the citadel and barbettes. From the ends of this 7-inch belt to the ends of the ship the armor is 5½ inches and 4 inches in thickness. A 'thwartship bulkhead of 6-inch armor is fitted at the after extremity of the water-line belt to afford protection against raking shot. On the advice of the Vickers Company, the Japanese Government, with their characteristic progressive tendency, have introduced in the *Mikasa* a departure which must be of interest to all naval constructors, the effect being to increase the area of the armored broadside. Above the main armor, and extending for the whole length of the citadel, there is 6-inch armor up to the level of the upper deck, instead of to the main deck as in previous ships, so that the total width is 21 feet 6 inches for the length of the citadel, and 7 feet 9 inches forward and abaft it.

The two barbettes are 14 inches in thickness, reduced to 10 inches where protected by 6-inch citadel armor; while the guns, as shown on the elevation, are protected by armored shields. The conning tower is of 14-inch armor, the screens being 12-inch. The whole of the armor, it is almost needless to say, is constructed on the Krupp principle, and is of a nickel-steel alloy.

The extension of the main belt to the upper, instead of to the main deck, as in preceding ships, dispenses with the necessity for casemates for the 6-inch guns on the main deck; but to insure the same measure of isolation for the ten 6-inch quick-firing guns placed on this deck as is obtained by the casemate system, bulkheads are placed between the guns, while an armored screen extends the whole length of the citadel behind the guns, as shown on the half plan of the main deck. These divisional bulkheads and the armored screen are of hardened steel—sufficient to insure that the splinters of any shell which may explode in one compartment will not affect either the gun or gunners in the adjoining compartment. The 6-inch guns on the upper deck, four in number, are within 6-inch casemates.

The protective deck is of a minimum thickness of 2 inches, and extends over the full length of the ship; but is increased to 3 inches on the sloping parts within the citadel, andto 4½ inches

in thickness in the wake of the barbettes, as before mentioned. In addition to this, the upper deck within the citadel is made of 1-inch plates. As usual, there is an ammunition passage under the protective deck with communication, through ammunition tubes, to the various gun positions; while coal bunkers are arranged above and below the protective deck from end to end of the citadel. There is thus an increased broadside protection to the citadel, as well as in the ends of the ship.

The general dimensions of the *Mikasa* do not differ materially from those of the British battleships of recent type. The ships of the *Majestic* and *Canopus* class were 390 feet long, the *Formidable*, 400 feet long, and the *Duncan*, 405 feet long—all between perpendiculars. The *Mikasa* is 400 feet, the length over all being 432 feet. The breadth molded is 76 feet in the *Mikasa*, as compared with 75 feet 6 inches in the *Duncan*, and 75 feet in the *Formidable* and *Majestic*. The draught is 27 feet 2 inches, and at this the displacement will be about 15,200 tons.

Like the two preceding ships for the Japanese navy built in this country—the *Asahi* and the *Hatsuse*—the *Mikasa* carries four 12-inch breechloading guns fitted in pairs in the two barbettes on the upper deck, and fourteen 6-inch quick-firers, four placed, as already indicated, within casemates situated on the upper deck at the corners of the citadel, the remaining ten 6-inch guns being placed in the box batteries on the main deck. The plans given show the radii of the respective guns. The auxiliary armament of the ship consists of twenty 12-pounder quick-firing guns, four firing ahead and four astern, the remainder being on the broadsides; eight 3-pounder quick-firing guns and four 2½-pounder quick-firing guns. There are four submerged torpedo tubes, two in the forward part of the ship and two aft.

In her electric and compressed-air equipment she embodies the best modern practice, the lighting installation including 900 incandescent lamps, with masthead, bow, anchor and other special lights, and also six searchlights. The ship will be fitted as a flagship, accommodation being provided for an admiral and 78 officers, the total complement being 935 all told.

The *Mikasa* is to have a speed of 18 knots under natural draft, and will have a coal capacity of 1,400 tons, to give her a radius of action of 9,000 sea miles at 10 knots speed. The engines are of the triple-expansion 3-cylinder type, the diameter of the cylinders being 31 inches, 50 inches, and 82 inches respectively, the stroke being 48 inches; and it is anticipated that the full power of 15,000 indicated horsepower will be developed with the engines running at 120 revolutions per minute, the steam pressure at the engines being 250 pounds, and at the boilers 300 pounds. Stephenson link motion is adopted for working the valves, which are of the piston type on the high-pressure and intermediate cylinders, and of the double-ported flat design on the low-pressure cylinders. The engines are designed to run the propellers inwards when going ahead, so that the starting platform is in the center of the ship, and here, as is now the universal practice, wrought-steel columns form the front supports of the cylinders, which are independent castings.

The back supports of the cylinders are of the ordinary cast-iron A-framing, with ample slipper-guide surface. The condensers are separate, and placed in the wings of the ship. The total cooling surface is 16,014 square feet.

The bedplate is of cast steel. The working parts of the engine generally are of forged steel, but the crank shaft is of nickel-steel, the three parts being interchangeable. The external diameter is 16¼ inches, with an 8¼-inch hole. The propeller shaft is of wrought steel 16½ inches in diameter, with 8½-inch hole, while the propellers have four blades, the diameter being 17 feet, and the mean pitch 18 feet. The four blades and the boss are of manganese bronze.

The boilers are of the Belleville type, with economizers. There are 25 in all, of which 20 consist of eight elements of generating tubes and seven elements of economizer tubes; while the remaining five boilers have seven elements of heating and six elements of economizer tubes. The total heating surface is 37,452 square feet, and the grate area 1,276 square feet. The two funnels are 14 feet in diameter over the casings, the height from the firegrate being 91 feet.

There is the usual complete system of pumps, with blowing engines for the furnaces. The total weight of the machinery is 1,355 tons. The machinery is practically completed, and ready to be placed on board the ship, which is to be completed for sea in May of next year.

The launching ceremony, it may be added, was conducted by Baroness Hayashi, wife of Baron Hayashi, the Minister in this country for the Emperor of Japan.—"Engineering."

RUSSIA.

Retvizan.—This battleship, built by the Wm. Cramp & Sons, for the Russian government, was launched at the Philadelphia ship yard, October 23, 1900.

The *Retvizan* is 384 feet in length on the water line. Her breadth is 72 feet 2½ inches, and the depth to main deck 43 feet 10 inches. At 25 feet draught, her displacement in tons will be 12,800, which is about the same as that of the new battleships *Maine*, *Missouri* and *Ohio*. The guaranteed speed is 18 knots. The main belt of armor along the sides of the *Retvizan* will be 8 inches in thickness, treated by the Krupp process. Above the belt the armor will be 6 inches thick. On the protective deck the flat surfaces will be 2 inches thick, while the slopes will be 3 inches in thickness. The motive power will be in two triple-expansion, vertical, direct-acting engines, placed in separate watertight compartments, and supplied with steam by twenty-four water-tube boilers of the Niclausse type, built in this country. The pressure will be 256 pounds. The bunker capacity is 2,000 tons. The *Retvizan* will carry two turrets, one placed at each end of the central superstructure, in which will be mounted four 12-inch breech-loading rifles. Included in the main battery also will be twelve 6-inch rapid-fire guns. The secondary battery will consist of twenty 75-millimeter rapid-fire, twenty 47-millimeter rapid-fire, and six 37-millimeter rapid-fire guns. There will also be two under-water torpedo tubes as well as four above water.

Accommodations for officers and crew will be ample and thoroughly well arranged. On the *Retvizan* thirty-eight officers will

be stationed, as well as 700 men. The electric installation throughout the craft will be ample and complete, and a number of novelties in this particular department are being introduced. All wood work in the interior of the vessel will be rendered fireproof by the electric fireproof process. As the modern warship depends so much upon electric power for the working of her interior mechanism, each new vessel shows some advantage over her predecessor in the same class. The *Retvizan* will be well supplied with electrical apparatus for almost every use except for propelling purposes.

Bayan.—The imperial Russian cruiser *Bayan*, which was recently launched at La Seyne (as noted in the JOURNAL of August last), is in some particulars what she claims to be—a new type of armored cruiser. She is novel in that many of her 12-pounders are behind armor, and she is novel also in the class of armor she carries. The general dimensions and features of the *Bayan* are as follows:

Displacement, 7,800 tons; length, 443 feet; beam, 55¾ feet; draught (mean), 22 feet. Guns (all quick-firers), two 8-inch; eight 6-inch, 45-calibers; twenty 3-inch, 12-pounders; seven 3-pounders. Torpedo tubes: Two submerged; three above water.

The guns are all Russian, from Obuchoff; the torpedo tubes in the case of those submerged will probably be the Elswick patent, which La Seyne has already supplied to some of its ships. The armoring is very extensive, and English influence is particularly noticeable in it. Indeed, but for the amidships redoubt and the absence of double casements, the plan might at first sight pass for that of the British *Essex* class. There is nothing characteristically French about it, though it is fairly well in line with recent French construction and design.

The armor is Krupp. The belt, which reaches from the bow to the base of the mainmast, varies from 8 inches to 4 inches in thickness and is not complete as in French design. It is associated with a curved protective deck, a form of protection essentially French. The deck is 2 inches thick on the slope. The belt is about 360 feet long. Above the water-line belt is a belt

of thin armor protecting the lower deck. This is three inches thick and extends 335 feet from the extreme bow. It will just keep out shell from 6-inch guns, but not shot. This belt protects the lower deck and two of the three above-water torpedo tubes. The third tube, in the stern, is unprotected. This upper belt covers the lower deck and reaches to the main deck, upon which all the 6-inch guns are carried. On this main deck there are three separate redoubts, all three inches in thickness. The forward and after ones both contain a pair of 6-inch guns, one on either side. The central and large redoubt contains four 6-inch guns and also eight 3-inch 12-pounders, carried four each side. This is the novel feature. It is not clear what isolation will be given to these guns in the way of screens; probably, bearing in mind the size of the ship, it will not be very much. Above each 6-inch gun 12-pounders are carried behind shields. The remaining four are on the main deck—two well forward and two right aft. On the high forecastle, and aft on the top of the main deck, are the 8-inch guns in closed balanced turrets, 7 inches thick. The hoists of these are 3 inches thick. There is also a 3-inch tube protecting the voice pipes from the conning tower, which is 6½ inches thick and placed up rather high.

All Russian ships carry plenty of search lights. The *Bayan* has one in the foretop, one carried low on the mainmast, one in the bow under the forecastle, one in the stern and one each side amidships, carried on top of the central battery. There will be four large funnels serving twenty-six Belleville boilers, fitted with economizers and placed in four groups. The stokeholds are eight in number, and each contains three boilers, except the aftermost pair, which contain four each. The engines are designed to work up to 17,000 I.H.P., with an estimated speed of 21 knots, which, according to contract, has to be maintained for twelve hours. The coal carried is 750 tons, but there is stowage room for 1,100 tons. This—as with water-tube boilers the consumption for all purposes is approximately a ton a knot—gives the ship a full-speed radius of 1,300 miles. It will also work out about 3,000 miles at 18 knots. Theoretically, of course, the radius is larger, but theory generally omits an

important item of consumption—auxiliary purposes. The *Bayan* has twin screws. She is a cruiser, pure and simple, built with an eye to cruiser work. The modern cruiser must carry vertical armor if she is to survive a single action. This naval officers have long and always contended, though ship designers have taken some time to realize it. The quick-firer left no alternative and it is probably folly nowadays to lay down any ship, no matter what her size, that is devoid of some sort of armor protection of the vertical sort. Just as no one now dreams of building cruisers without protective decks, so, by-and-by, no one will dream of building them unarmored, no matter what their class. This is the general trend of naval opinion in all nations; and there will probably be no mean between the vertically-armored ship and the unarmored destroyer.—Exchange.

MERCHANT STEAMERS.

Morro Castle.—The American steamship *Morro Castle*, built for the "Ward Line" by the William Cramp & Sons' S. and E. B. Co., Philadelphia, left on her first voyage, on November 10, 1900, from New York to Havana, and reached her destination on November 13th, making the trip in three days and four hours, and breaking the record between these ports.

This is one of the finest steamships built in this country for coasting service, and has been constructed under the Act of March 3, 1891, which provides for Government inspection and requires certain characteristics which will enable these ships to be readily converted into auxiliary cruisers in time of war. This vessel successfully passed the required inspection for vessels of the 16-knot class, and on her official trial, July 14th, 15th, made an average of 17.57 knots per hour. Since that time, and just prior to her entering regular service, further tests were made by a board of naval officers to ascertain the maximum speed which could be maintained at sea under ordinary conditions of wind and weather. The speed was noted from the revolutions of the engine, the screws standardized by preliminary runs over a measured course, and demonstrated the ability to maintain a speed of 17.74 knots. This average was held for 17 hours, when the ship returned to the Delaware River.

The following is a general description of the vessel, with general data of tests:

Length on load-water line, 400 feet; breadth, molded, 50 feet; depth, 36 feet 6 inches; gross registered tonnage, 6,004 tons; net tonnage, 3,732 tons; displacement of load draught, 8,280 tons; deep-load draught, 21 feet; light draught forward, 8 feet 10 inches; light draught aft, 15 feet 7 inches; light draught mean, 12 feet 2½ inches; capacity of bunkers, 915 tons.

The hull is subdivided by six watertight compartments. In the construction of the hull only the best tested steel has been

used. The frames are of bulb angle steel, 7 inches by 3½ inches by 20 pounds, carried up alternately to the hurricane and main decks from stern to within 60 feet of stem, forward of which all frames are carried to hurricane deck. The frame spacing is 24 inches. The main, lower and orlop decks are completely of steel, while the hurricane and shade decks are of wood with steel stringers, tie plates and margin plates. The vessel has a double bottom for nearly her entire length, which not only affords additional protection in case of grounding, but can be utilized for carrying water ballast or a fresh water supply for the boilers.

The capacities of the trimming and double-bottom tanks are as follows, when filled with salt water: Forward trimming tank, 121 tons; after trimming tank, 165 tons; double-bottom compartments, 746 tons; total, 1,032 tons.

The passenger accommodations are roomy and well ventilated and provide for 136 first-class, 60 second-class, and 44 steerage. The closets, lavatories, bathrooms, etc., for passengers and crew are very ample in number and splendidly arranged.

The vessel is propelled by twin screws, driven by two vertical 4-cylinder triple-expansion engines, making about 8,000 I.H.P., at 100 revolutions, with a working steam pressure of 170 pounds. The cranks are set at angles of 90 degrees. The cylinders are as follows: High-pressure cylinder, 32 inches; intermediate-pressure, 52 inches; two low pressure, each 60 inches; number of revolutions per minute, 100; condensers, cooling surface, 6,000 square feet.

The propellers are of the 3-bladed, solid type, of Parsons manganese-bronze, of the modified Griffith type. The diameter of each is 14 feet, with 22 feet 9 inch pitch.

The main shafting is of mild steel, the propeller shafts being 14½ inches in diameter, intermediate shafts 13¾ inches, and thrust shafts, 13¾ inches.

There are eight main boilers, of the single-ended, return-tubular, cylindrical marine type, each 14 feet 2⅝ inches in diameter by 10 feet 5½ inches. There are three Morison corrugated furnaces in each boiler, the furnaces being 39 inches in diameter.

The working pressure is 170 pounds. The total grate surface is 442 square feet, and the total heating surface, 18,200 square feet. For forced draft there are four Sturtevant fans, each 66 inches in diameter, width at periphery, 18 inches, and revolutions per minute, 480. The smoke pipes, two in number, are each 7 feet 9 inches inside diameter. The donkey boiler is 10 feet long by 10 feet in diameter.

The boilers are below the load-water line and therefore are well protected, and the upper ends of the cylinders, which are above the water line, are protected by coal.

As an auxiliary cruiser in time of war the *Morro Castle* would prove an invaluable addition to our navy on account of her size, her speed and strength, for she is exceptionally well and strongly built. Special strengthening and foundations are provided to carry three 6-inch guns on the hurricane deck, and eight 6-inch guns on the main deck. The arrangements of the upper decks are such that an efficient secondary battery can be easily installed.

FULL-POWER CONTINUED SEA SPEED TRIAL, OCTOBER 12 AND 13, 1900.

Average total I.H.P. main engines without auxiliaries	9,415	
Estimated H.P. with auxiliaries	9,886	
	Port.	*Starboard.*
Revolutions	106.1	105.9
Steam pressure	172.4	172.4
First receiver	62.3	61.1
Second receiver	12.6	12.9
Vacuum	25.24	25.1
Double strokes air pump, port cylinder	39.59	43.5
starboard cylinder	38.97	42.7
Revolutions circulating pump	131.	133.8
Temperature, injection	68.	65.7
discharge	121.	131.
hot well	122.	
feed	205.	
Grate surface in use, square feet	442.	
H.P. per square foot grate surface, main engines	21.28	
total	22.35	
Heating surface in use, square feet	18,203.	
per H.P. main engines, square feet	1.93	
total, square feet	1.84	
Average speed, knots per hour	17.74	
Duration of run	16 h. 56 m. 30 sec.	

This run was, of course, held under forced draft, for which this machinery is particularly designed. The air pressure averaged in the ash pans from 1.4 to 2.3 inches of water.

The temperatures of the base of the stacks averaged 378 degrees.

The consumption of coal at 10 knots speed is but 30 tons per day, giving a radius of action of over 7,000 miles.

St. Paul.—The steamship *St. Paul* met with a serious accident on October 31, while on her way from Southampton to New York. The starboard propeller shaft parted and the propeller was lost. As the ship was pitching heavily at the time, it is thought that the break was due to the racing of the engine. As a result of the break the engine was badly wrecked. Steam was shut off by a heroic member of the engine-room force before injury was done to the hull of the vessel or the longitudinal bulkhead, and the ship proceeded to port with the remaining engine.

The propeller shaft broke off about 9 feet inside the stern tube, and this sudden release of resistance caused such an acceleration of speed to the reciprocating parts of the engine as was beyond the strength of the cap bolts of the main bearings, together with the "cushioning," to retard at the upper centers of the forward engines. Here the weights of moving parts were greatest, as the two H.P. cylinders are directly over the two great L.P. cylinders, in tandem fashion, working on the two forward cranks. The result was that instantly the cap bolts of the forward crankshaft bearings stripped their lower threads, the crank shaft itself was raised some 4 inches from its bed, the H.P. and L.P. pistons striking their upper cylinder heads, broke these four heads completely off, throwing the lighter H.P. covers over into the upper grating, smashing one crosshead slipper, bending the after L.P. piston rod and cracking the L.P. bed-plate castings themselves beyond repair. The raising of the forward end of the crank shaft "drew" the crank pin of the second L.P. crank shaft perceptibly in its eye in the crank, and probably bent the shaft, also.

The after main bearings did not appear to suffer and the tun-

nel shaft and thrust bearings were intact. The marvel was to find such a complete wrecking of the forward cylinders without any fatality attending the causualty, and without actually imperiling the ship more seriously.

Runic.—The White Star steamship *Runic* was launched at Belfast on October 25. This vessel is 565 feet long, 64 feet beam and 12,400 tons gross tonnage. She has twin screws and is generally similar to the *Afric*, *Medic* and *Persic*, of the same line. In this ship the dining room, seating 400 passengers, and the writing and reading rooms are all on the upper deck. The *Runic* can carry 100,000 carcasses of mutton in refrigerated compartments, and 20,000 bales of wool. She is intended for the South African and Australian service.

Ernest Woermann.—Messrs. J. Dunlop & Co., Glasgow, launched, on August 14th, the *Ernest Woermann*, the first of two large passenger and cargo steamers building for the Woermann Line of Hamburg. The principal dimensions are: Length, 540 feet; breadth molded, 44 feet; depth molded to main deck, 29 feet 6 in. A full poop extends 230 feet from aft, and there is a topgallant forecastle, 44 feet long. The other erections consist of a promenade deck aft for second-class passengers, poop deck 60 feet long, and a similar promenade deck for first-class passengers, carried from fore part of long fall poop for a distance of 114 feet, whilst over this again is carried a boat deck also 114 feet, on the fore end of which is situated the captain's room and chart room, with a navigating bridge over same and extending to ship's side at a total height of 62 feet from keel. The erections above the line of poop and forecastle decks are all open at sides, having efficient and strong supporting and protecting stanchions and rails as necessary in such vessels. The propelling machinery consists of one set of triple-expansion engines, the diameter of the cylinders being 24 inches, 38 inches and 65 inches, with 42-inch stroke. Steam is supplied by natural draft by three large single-ended Scotch boilers working at a pressure of 180 pounds to the square inch.

Noranmore.—The turret steamer *Noranmore* arrived at Baltimore on October 26, from Liverpool. This is the largest turret

steamship afloat and belongs to the Johnston Line. The *Noranmore* is built of steel and is 420 feet long, 50 feet beam, and carries a cargo of 9,000 tons. She has triple-expansion engines of 27½, 45½ and 75 inches diameter of cylinders. When loaded she has a free board of only 16½ inches, with a bridge 35 feet above the water surface. She is a modification of the American whale back built by W. Doxford & Son, of Sunderland, and possesses the merit of carrying a cargo of 9,000 tons on a registered tonnage of 3,650 tons, and 5,600 tons gross. She is said to be a good sea boat and dry.

Hardinge.—The Fairfield Shipbuilding and Engineering Co., Limited, launched on August 11th the Royal Indian Marine steamer *Hardinge*, built to the order of the Secretary of State for India in Council, for the service of the Government of India. The principal dimensions are: Length, 423 feet 6 inches; breadth, 51 feet; depth, 31 feet; gross tonnage, 5,600 tons; boiler pressure, 180 pounds, and speed, fully loaded, 18 knots. The propelling machinery consists of two sets of triple-expansion surface-condensing engines. The cylinders are 29 inches, 46 inches and 72 inches in diameter, with 4 feet stroke. The boilers for generating the steam for the engines are five in number, of the ordinary multitubular marine type, and arranged to work with forced draft. Each of the double-ended boilers has six furnaces, and the single-ended three, making a total of 27 furnaces, the products of combustion being led into two funnels. All the work has been done under the supervision of Sir E. J. Reed, K. C. B., naval architect and engineer to the India Office, with Mr. F. F. Hill as resident overseer.

Pennsylvania.—An unusually interesting vessel, the *Pennsylvania*, was recently put into service on the waters of Chesapeake Bay in what is generally known as the Cape Charles Route.

The new steamer is a handsome, shipshape vessel, with the usual American steamboat superstructure, two well proportioned funnels and a pleasing amount of sheer. Her chief point of peculiarity, or difference from other vessels engaged in similar service, is that she has a modified torpedo-boat or knuckle stern. The stern of the *Pennsylvania* differs from the regular torpedo-boat

stern in that it has practically two sterns divided at the knuckle, in the same style as the English channel steamer *Tamise*, of 1,000 tons.

The *Pennsylvania* was designed by Naval Architects Gardner & Cox, of New York, and built at the Roach yard, Chester, Pa. Her principal dimensions are: Length over all, 260 feet; between perpendiculars, 247 feet; beam molded, 37 feet 8 inches; beam over guards, 40 feet 9 inches; depth molded at center, 15 feet; depth molded at side, 14 feet; depth of hold, 13 feet; draught, 9 feet 6 inches; displacement, about 1,100 tons.

Boats carried include four 24-foot metallic life boats, and there is the usual supply of life rafts and life preservers. On each side of the vessel there are three large freight ports about 8 feet square, between the main and saloon decks, and the passengers' gangways are located opposite the main lobby on the main deck.

Propelling engines of the *Pennsylvania* driving the twin screws are of the vertical, four-cylinder, triple-expansion type, with cylinders 18 inches, 29 inches and (2) 33 inches diameter and 26-inch stroke. All the main valves are of the piston type, worked by Stephenson gear with double-bar links. The framing for each engine consists of six pairs of cast-steel columns, well stiffened by diagonal ties. Cast-iron is used for the bed plates, which are carried by wrought-steel keelson plates forming the foundation. Each crank shaft is in two sections, and all shafting, piston and connecting rods and working parts of the valve gear are of forged steel. Each engine was designed to indicate 1,500 I.H.P. at 180 revolutions, with 200 pounds steam pressure.

Special interest attaches to the boiler equipment of the *Pennsylvania*, as we believe this is the largest installation of water-tube boilers of the "small tube" variety in a passenger vessel in American waters. There are six water-tube boilers of the *Almy* type, built to work under forced draft at a pressure of 225 pounds per square inch. They are fitted forward of the engines in two sets of three abreast, and fronting on a common athwartships fire room. Each boiler occupies a space about eight feet square, and the fire room is twelve feet long in a fore-and-aft direction

and 25 feet wide. Each group of boilers is connected to its own stack.

The total grate surface is 363 square feet, and the total heating surface 10,960 square feet. The weight of the water-tube boilers, filled with water to the steam level and including all furnishings except the breechings and smokestacks, is 76½ gross tons.

All auxiliaries are independent. There are two independent surface condensers, each with a centrifugal circulating pump, two vertical beam air pumps, two vertical duplex feed pumps, and also the usual ballast, bilge and sanitary pumps. Steam-steering engine is also fitted, and forward there are two steam lifts on the main deck for hoisting freight in and out of the hold, and there is also the usual equipment of steam capstan and windlass. A very complete electric-lighting plant also forms part of the equipment.

The *Pennsylvania* was designed to meet the contract conditions which stipulated that she should be capable of steaming at the rate of 20 statute miles per houron a draught of 9 feet 6 inches when carrying 300 tons of freight, besides the coal, water, stores, and passengers.

Deutschland.—The following more complete description of the Hamburg-American Line twin-screw steamer *Deutschland* is taken from the "Marine Review" and materially augments the data previously published by the JOURNAL on this vessel. The principal dimensions are: Length, 662 feet 9 inches; length over all, 686 feet 6 inches; breadth, 67 feet, and depth molded, 44 feet. The net registered tonnage is 16,000, and the load displacement about 23,000 tons. The vessel's rig is of the lightest description, there being only two pole masts. The rudder is totally submerged and the propellers are carried close in and well under the hull, an aperture being provided as in single-screw propelled vessels. Spare steering gear is fitted below the water line. The launching weight of the hull exceeded 9,200 tons. There are four funnels.

The boilers—twelve double and four single-ended—having altogether 112 furnaces, being placed in four water-tight com-

partments, a fifth and central compartment being devoted to a large cross coal bunker, so causing the greater space between the second and third funnel. The double-ended boilers are 20 feet 4 inches long, 16 feet in diameter, each of the four furnaces at each end of the boiler, which are Morison's patent, being 3 feet 6 inches in diameter. The total grate surface of the boilers is 2,200 square feet, and the total heating surface, 84,250 square feet. Howden's system of forced draft is adopted, the tubes, which are 3 inches diameter, being fitted with retarders. Four of the ballast tanks under the boilers are set apart for carrying about 300 tons of fresh water for feeding the boilers. There are two Blake simplex pumps capable of pumping from the sea, the bilges and the double bottoms, and discharing overboard or into the fire mains. Each of the four boiler sections contains one Blake vertical duplex feed pump, size 12 inches by 8½ inches by 12 inches, capable of feeding the boilers of that section. The pumps have connections which enable them to draw water from the sea, fresh-water tank, from the boilers and from the bilges, and discharge overboard or into the boilers. Water to the sanitary system, is supplied by two Blake vertical, simplex pumps, size twelve inches by 12 inches by 16 inches, and also by two pumps directly connected to the main engines.

The engines are of the six-cylinder, four-crank, quadruple-expansion type. The cylinders have a common stroke of 1,850 mm. (72.8 inches), and the diameters of the cylinders are as follows: Two high-pressure cylinders, 930 mm. (30.6 inches); first intermediate-pressure cylinder, 1,870 mm. (73.6 inches); second intermediate-pressure cylinder, 2,640 mm. (103.9 inches); two low-pressure cylinders, 2,700 mm. (106.3 inches). The two low-pressure cylinders are in the center, with the two high-pressure cylinders over them, and at the forward end is the first intermediate and at the after end the second intermediate. The two first cranks, set opposite each other, have thus the intermediate in the one case, and a high-pressure and low-pressure cylinder in the other, and the after pair of cranks, a high-pressure and low-pressure in the one case, and the intermediate on the other crank. The cylinders are placed close together, the valves

being on the outside, and there is a separate valve gear for each cylinder, i. e., six sets for the six cylinders of each engine. The end cylinders have their valve boxes on the outside, and, owing to the great diameter, there are two spindles. The crank shaft is of nickel-steel and is hollow, the outside diameter being 25.2 inches. The propeller shafts are also of nickel-steel, while the intermediate shafting is of Siemens-Martin steel. The thrust shafts are the same diameter as the crank shafts, and are placed immediately abaft the engines, each having twelve thrust collars. The thrust blocks are of the ordinary horseshoe pattern with white metal facings.

The engine frames are steel castings, sixteen in number for each engine, each frame receiving the horizontal thrust of a guide shoe from the crossheads. There are sixteen guide shoes, all of the same size, for each engine, lined with white metal, four to each crosshead, two for going ahead and two for going astern. The intermediate pistons are fitted with tail rods. The high-pressure pistons are fitted with Ramsbottom rings, the intermediate with Buckley's rings and the low pressure with Peck's patented rings. The valves are worked by ordinary Stephenson double-bar links. The high-pressure valves obtain their motion from a rocker shaft which is placed upon the low-pressure cylinder covers. One end of the rocker arm is attached, by means of links, to the valve rods, and the other end to a link rod passing between the intermediate and low-pressure cylinders, and is attached to the valve gear below. The high-pressure cylinders each have one piston valve, and the intermediate pressure cylinders each two piston valves. Each low-pressure cylinder has a flat slide valve, operated by two valve rods connected to a yoke. All the valves, except those of high-pressure cylinders, are fitted with balance pistons. The rock shafts are placed on the outboard side of the engines, and are controlled by Brown's combination steam and hydraulic reversing engines, hand gear being also fitted. Each engine room contains an independent cylindrical condenser. The tubes are $\frac{3}{4}$-inch diameter, the cooling surface for each condenser being 21,315 square feet. There are two independent Blake air pumps of the vertical "twin"

type, double-acting in the steam ends, and with single-acting water ends. The steam cylinders are 18-inch diameter, water ends, 44-inch diameter, with a common stroke of 24 inches.

Cooling water to the surface condensers is supplied by two centrifugal pumps, both driven jointly by one compound engine. These pumps can also draw from the bilges. The discharge of the circulating water from the condenser is under the water line. Two double, Weir feed pumps are fitted in each engine room, having steam cylinders 12-inch diameter, and water cylinders 8-inch diameter and 12-inch stroke. The inboard feed pumps draw from the hot well and deliver the water into a Weir feed-water heater. The outboard pumps take the water from the heater and, delivering the water through a Claussen oil-extractor, pump into the boiler-feed line. Besides these connections the feed pumps can also draw from the fresh-water tanks and pump directly into the boilers or overboard. The steam used for heating the feed water is taken from the second-intermediate receiver and the auxiliary exhaust.

Morison's radial-type evaporators, capable of delivering about 23,770 gallons per diem, is fitted, as well as an auxiliary condenser for use in port. There are two four-bladed propellers, each 23 feet diameter, the pitch being adjustable from 33 feet to 36 feet. The total power of main engines at 78 revolutions a minute, about the normal full speed, is 36,000 I.H.P., although on the maiden trip 36,600 I.H.P. was recorded. The owners only expected 35,000 I.H.P., with an average speed of 23 knots. The vessel has thus come up to, and even exceeded their anticipations, as on the second eastward passage the average speed was 23.36 knots, and on the first passage eastward, 23 knots. Being a twin-screw steamship, a central longitudinal bulkhead has been adopted, so facilitating the sub-division of the hull as to make the vessel, we understand, practically unsinkable, it being divided into seventeen watertight compartments, besides the double bottom, which is sub-divided into twenty-four tanks. The bulkheads have been constructed in accordance with the regulations of the Germanischer Lloyd and the German Government, and are considered to be sufficiently strong to resist the greatest pressure

of water they could be subjected to. Not only has the safety of the vessel been considered in the design, but also the comfort of the passengers, for which purpose, amongst other arrangements, bilge keels have been fitted.

The vessel has accommodation for 467 first-cabin, 300 second-cabin, and 300 third-class passengers. There is provided for the first-cabin passengers, in addition to the large dining saloon, ladies' parlor, and smoking room, a play room for children, and a gymnasium and a grill room on the boat deck. The total length of the promenade deck for saloon passengers exceeds 550 feet, and, in addition, the boat deck is at their disposal for promenading. The second-cabin accommodations are located aft on the promenade, upper, main and lower decks.

Great attention has been paid to the sanitary arrangements, the bath rooms being thirty-three in number, of which sixteen are first-class, six for the suites, six second class, one for the captain, two for the engineers, and two for the hospitals. The electric-lighting plant consists of five direct-connected dynamos, of which three are of 700 ampères and 110 volts. The two-wire system is used. There are about 2,000 lamps in the circuit. The refrigerating machinery is placed between the two propeller shafts, and consists of carbonic-acid gas machines. The number of life boats is twenty-six, of which eighteen are metal life boats of the Francis' patent, two are wooden, and six are of the collapsible type. The vessel, besides having the ordinary mechanical telegraphic connections, is fitted with a complete system of telephonic connections, thus enabling the officers on the bridge to be in verbal communication with the engine-room force and the steering-engine-room crew.

The *Deutschland*, as well as the rival steamer of the Norddeutscher Lloyd, *Kaiser Wilhelm der Grosse*, which she has supplanted, was built by the Stettiner Mischinenbau Actien Gesellschaft "Vulcan," of Stettin, who have on hand another new steamer, to be named the *Kaiser Wilhelm II*, to excel the performances of the *Deutschland*, as well as a steamer to equal the *Kaiser Wilhelm der Grosse*, to be called the *Kronprinz Friedrich Wilhelm*.

YACHTS.

Victoria and Albert.—The new royal yacht, *Victoria and Albert*, has now completed her steam trials (August 9 to 26, 1900), and the official results are given in the appended table, which shows that in all respects the expectations have been more than realized. The yacht was designed by Sir William H. White, K. C. B., the director of naval construction, and was built at the Pembroke dockyard. She is 380 feet long by 50 feet beam, and her displacement is 4,700 tons, with 350 tons of coal in the bunkers. Great importance is attached to her stability, and the stormy weather experienced afforded opportunity of testing it, and on all occasions with satisfactory results.

The engines are twin-screw, four-cylinder, triple-expansion type. The high-pressure cylinder is 26½ inches in diameter, the intermediate, 44½ inches, and two low-pressure cylinders, 53 inches, with a stroke of 3 feet 3 inches. The boilers are of the Belleville type, with economizers, and have 26,000 square feet of heating surface and 840 square feet of grate area. The stokeholds are artificially ventilated, there being no cowls. The machinery, it may be added, was designed and constructed under the special direction of Sir John Durston, K.C.B., Engineer-in-Chief of the Navy.

RESULTS OF CONTRACT STEAM TRIALS OF THE NEW ROYAL YACHT *VICTORIA AND ALBERT.*

—	48 hours at 5,000 I.H.P.	46 hours at 7,500 I.H.P.	48 hours at 7,500 I.H.P.	8 hours at full power.
Draught of water, forward	18 ft. 2 in.	18 ft. 2½ in.	18 ft. 2½ in.	18 ft. 2 in.
Draught of water, aft	20 ft. 1 in.	20 ft. 0 in.	19 ft. 11½ in	20 ft. 1 in.
Steam pressure, pounds	254	268	272	306
Vacuum, inches	26.9	26.0	25.25	25.25
Revolutions	110.9	128.4	128.4	147.4
Indicated horsepower	5,142	7,625	7,649	11,298
Speed over long course, knots	16.32	18.33	18.47	20.53
Coal consumption, pounds	1.94 *	1.94 *	1.87 *	—
Dates, August 1900	9 to 11	16 to 18	23 to 25	29

* For 40 hours.

The steam trials set by contract consisted of four runs, the first of 48 hours' duration, the next two, each of 48 hours' duration, at 7,500 indicated horsepower, and the fourth, of eight hours, at full power. The results of all four are tabulated above.

Any doubts entertained as to stability have been entirely dissipated by the behavior of the yacht in two gales, when her speed did not fall off to an appreciable degree, while in a moderate breeze she was perfectly steady. Whether she has had to steam through a heavy sea or through smooth water, the bow wave has been clean and regular, with a fine run from the stem to the midship section, and no volumes of water of weight have been thrown off from any part. Nor has there been any churning or thumping under the quarter, whether the ship has been traveling at her highest speed or running through a gale. Each trial also disclosed an absence of vibration over the propellers, while the tests that were made in the worst weather she encountered showed that the angle of heel in rolling did not exceed 8 degrees or 10 degrees. The hull and engines have thus answered all the requirements of the designers.—" Engineering."

Lysistrata.—The large, full-powered, sea-going steam yacht *Lysistrata*, for James Gordon Bennett, owner of the " New York Herald," was launched from the Denny yard, at Dumbarton, on August 28. This vessel is of the following dimensions: Length on water line, 285 feet; length over all, 314.5 feet; breadth, extreme, 39.9 feet; Thames tonnage, 2,082. She has a straight stem and long stern, and there is a full poop aft, a long shade deck amidships, and a short turtle back forward. Unlike other large yachts, she has only one mast, which is placed aft of the funnel, and has a short yard for signaling purposes only.

The machinery of the yacht consists of two sets of four-crank triple-expansion engines, and four single-ended boilers of the cylindrical multi-tubular type, with a donkey boiler of the same type. She has ample coal bunkers, and it is intended she should attain a high speed. She will be lit throughout with electric light.—" Marine Engineering."

BOOK REVIEWS.

THE MONITOR AND THE NAVY UNDER STEAM, BY FRANK M. BENNETT, LIEUTENANT, U. S. NAVY.—Lieutenant Bennett has furnished the publishers of the above volume fitting material for an example of elegant book work, and the publishers have done their part in presenting this story of the advance of steam in the marine world, and particularly in the navy, in a form as pleasing to the eye as the story itself is to the mind.

The author's style is particularly felicitous, and the reader is carried forward through the history of the marine steam engine without an abatement of interest by a single dull page.

Preceding the account of the production of the *Monitor* and *Merrimac* (both of which events are given with extended detail and careful illustrations), a comprehensive resumé of the signs and advance of earlier steam machinery, form a most valuable chapter, while the effect produced by the success of this new type of war ship is the keynote of the balance of the book, and includes the evolution of the battleship of to-day, with historical accounts of the important naval incidents down to the end of the recent Spanish war.

The publishers are Houghton, Mifflin & Co., Boston, Mass.

THE FRIGATE CONSTITUTION.—Ira N. Hollis, formerly a conspicuously brilliant member of the Engineer Corps of the U. S. Navy, and now at the head of the Engineering School at Harvard University, has told the story of the central figure in our conceptions of what the Navy was a hundred years ago, the story of the frigate *Constitution*, in a manner and with a completeness that can not fail to deeply interest the reader.

With the disappearance of the sailing ship, naval romance vanished, giving place to an unpoetic, though more continuously

energetic, mechanical phase of sea life and warfare. Yet few naval men can look at the picture of the old *Constitution* or read the history of her prowess, without feeling at once admiration for the beauty and grace of a full-rigged sailing ship and a regret that the days of harnessing the winds of heaven have departed.

Neither can one read of the deeds of those heroes who guided the old frigate through the period of her momentous and thrilling prime, without experiencing enthusiastic pride in their performances and the strongest desire to have that monument of American naval prestige perpetuated by rebuilding the ship, and preserving her original characteristics as an historical illustration that can not fail to ever awaken patriotic emotions.

The publisher has spared no pains to make the book attractive. It is profusely illustrated, and the typography is most excellent.

The publishers are Houghton, Mifflin & Co., Boston, Mass.

There are catalogs and catalogs, and frequently the display and press work are samples of high art, and the detailed information contained at least pictorially valuable, even if the reader does not know the discount, and has never heard of 80 and 10 off.

For many years the CATALOG OF MANNING, MAXWELL AND MOORE (85-7-9 Liberty street, New York), has held place at the top of the list of the great general machinery catalogs, and has been a book of reference at all our Yards and Stations.

We have just received a new edition of this work, comprising, however, only metal and wood-making machine tools and their attachments. This separation of the machine tools proper from the general supplies of this firm is most wise and opportune, as it relieves the catalog from the feature of cumbersomeness and at the same time facilitates the work of selection. Even in these specialties the new book contains 700 pages of Imperial quarto size, and is handsomely illustrated and bound.

For greater convenience in ordering goods, each article has a figure number, which may be used in telegraphic or written orders, instead of the tedious description of the article itself.

WESTINGHOUSE RAILWAY MOTORS.—Our attention has been called to a phamplet just published by the Westinghouse Companies Publishing Department which very attractively displays the several styles of Railway Motors now made by the Westinghouse Electric and Manufacturing Co. The use of half tone illustrations, direct reproductions from photographs of the details of the apparatus, not only adds greatly to the appearance of the phamphlet but renders the ready comprehension of the types shown far easier than would be possible from line drawings.

ASSOCIATION NOTES.

Naval Engineers will certainly congratulate the Westinghouse Electric and Mfg. Co., of Pittsburg, Pa., upon their fortune in securing the services of Mr. Asa M. Mattice as their Chief Engineer. We are informed that arrangements to this end have been concluded and that Mr. Mattice enters upon his new duties in December of this year.

The Engineer Corps of the Navy never included in its list of members a more capable, energetic and brilliant officer and engineer expert than Asa M. Mattice, and his work in the Bureau of Steam Engineering prior to his resignation in 1890, as assistant to Engineer-in-Chief Geo. W. Melville, was almost phenomenal in amount as well as importance.

After leaving the service Mr. Mattice was principal assistant to E. D. Leavitt of Cambridgeport, Mass., and for about ten years devoted himself to the designing of the large machinery with which the name of Leavitt has been so prominently connected. During the past year Mr. Mattice has been pursuing an independent line of work in remodelling the Cocheco Cotton Mills at Dover, N. H., and on completing this he accepted the offer made by the Westinghouse Electric and Mfg. Co., as noted above, and of which company his former Naval Academy classmate B. H. Warren, U. S. Navy, is Vice President.

A regular meeting of the Society was held at the Navy Department on October 2, 1900, for the purpose of nominating officers for the coming year. Nominations were made in writing, and resulted as follows:

For President: Commander Charles W. Rae, U. S. N., and Lieutenant-Commander F. H. Bailey, U. S. N.

For Secretary-Treasurer: Lieutenant B. C. Bryan, U. S. N., and Lieutenant-Commander A. B. Willits were nominated, but the latter declining, the nomination of Lieutenant Bryan was made unanimous.

For Council (three to be selected): Commander C. W. Rae, Lieutenant Commanders Jas. H. Perry and W. R. Worthington, Lieutenants R. S. Griffin and C. E. Rommell.

The vote will be secured by the usual voting slips, and will be counted at the regular meeting in January.

The matter of award for prize essays was discussed, and it was decided not to continue hereafter the system of voting for first and second prize for contributions. Endeavor will be made to secure important manuscript from members in a more direct and certain way, and the award for *second prize* will be discontinued entirely.